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AN EVALUATION OF STEADY-STATE DEHUMIDIFICATION CHARACTERISTICS OF RESIDENTIAL CENTRAL AIR CONDITIONERS

FINAL REPORT

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Energy Systems Laboratory
Department of Mechanical Engineering
Texas A&M University

Contract No. S-95013
Prepared for
Houston Lighting & Power
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CHAPTER 1
INTRODUCTION

Air conditioning systems typically perform two functions: cooling and dehumidification. In climates such as that in Houston, where summers are both warm and humid, the dehumidification capabilities of the air conditioner are of extreme importance for maintaining comfort in a building.

Some concern has been expressed in recent years about the inability of some "high efficiency" air conditioners to dehumidify properly in hot/humid climates. Since the oil embargo of 1973, and the passsage of legislation requiring development of mandatory minimum efficiency standards for new central air conditioners (and other appliances), manufacturers have been improving the efficiency of central air conditioners dramatically (Figure 1.1). Seasonal Energy Efficiency Ratios(SEERs (EERs before 1979)) have increased from 7.16 in 1976 to 8.84 in 1985[1]. These improvements in air conditioners have come about through improved compressor performance, higher efficiency motors, larger heat exchangers, improved heat transfer surfaces, and computer assisted system optimization. Two of the paths for improving air conditioners: larger heat exchangers and improved heat transfer surfaces, if not applied judiciously, can decrease dehumidification performance.

This report is the first of two reports on the project "Determination of the Transient Response Characteristics of High Efficiency Commercial Air Conditioners" funded by Houston Lighting and Power Company. The purpose of this report is to present the results from an investigation of the possible relationship between residential central air conditioner efficiency and dehumidification performance. The method used was that of a survey of air conditioners currently manufactured and on the market as of late 1986. Chapter 2 outlines the basic terminology used in evaluating the dehumidification performance of central air conditioners. Chapter 3 provides background to the survey data. Chapter 4 presents results from the survey. Conclusions and recommendations are presented in Chapter 5.

Reference

1. "Comparative Study of Energy Efficiency Ratios", Air Conditioning and Refrigeration Institute, Arlington, VA, 1986.

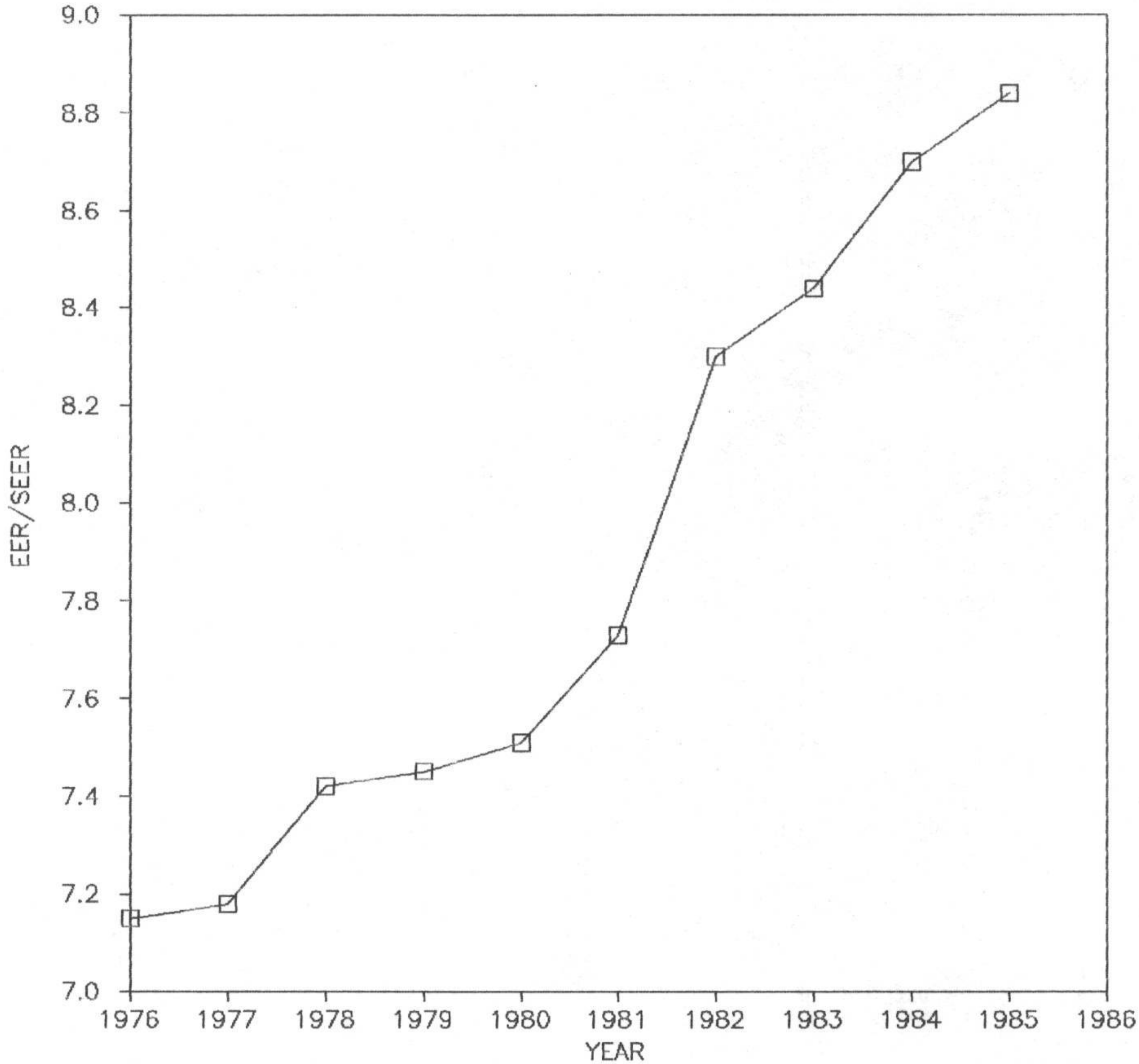

Figure 1.1 - Improvement in Split System Central Air Conditioners Over the Past Ten Years [1]

CHAPTER 2
COOLING/DEHUMIDIFICATION TERMINOLOGY

Before looking at the dehumidification performance of air conditioners, it is best to first define some of the terminology associated with residential air conditioners and the dehumidification process. Residential central air conditioners receive their nominal rating from the Air Conditioning and Refrigeration Institute (ARI). ARI certifies the capacity and efficiency of the unit. These data are available in the Directory of Unitary Air Conditioners, Air Source Heat Pumps, and Sound-Rated Outdoor Unitary Equipment that ARI publishes twice a year[1]. In addition to the ARI certification, many manufacturers publish detailed performance (capacity, efficiency, and dehumidification) data for their equipment operating at conditions other than those rated by ARI. This chapter outlines the terminology used in these data.

The data provided in the ARI Directory[1] require making four tests in test rooms where both humidity and temperature can be tightly controlled.* These rooms are called psychrometric rooms. The tests are referred to as tests A, B, C, and D[2]. Test A is a steady state test performed with an outdoor temperature of 95 F, and indoor conditions at 80 F dry bulb and 67 F wet bulb. The results of Test A are used as the advertised (or nominal) value of the capacity (in Btu/hr) and the steady state efficiency (Energy Efficiency Ratio) for the air conditioner. The EER of an air conditioner is defined as:

$$EER = \frac{\text{Cooling Capacity (Btu/hr)}}{\text{Power Input (watts)}} \qquad (2.1)$$

Tests B, C, and D are all run at an outdoor temperature of 82 F. Test B has the same indoor conditions as Test A. At the indoor conditions specified in both tests A and B, the indoor coil is dehumidifying the air. For Tests C and D, the indoor humidity is reduced so that the wet bulb temperature is 57 F (or less). At this condition, there is no condensation on the evaporator. Test C is a steady state test, while Test D is a cycling test. The unit is on for 6 minutes and off for 24 minutes. From tests C and D, the degradation in performance of the unit due to on-off cycling is estimated and used with data from Test B to calculate the Seasonal Energy Efficiency Ratio (SEER).

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*The test procedure used to rate air conditioners and heat pumps is reproduced in the appendix.

The SEER is defined as:

$$\text{SEER} = \text{EER}_B * (1 - 0.5 * C_D) \qquad (2.2)$$

where,

EER_B = energy efficiency ratio measured from Test B

C_D = cycling degradation factor established from Tests C and D

While the amount of dehumidification that an air conditioner provides is measured during both Tests A and B, ARI does not require manufacturers to report the results. All they are required to report in the ARI Directory[1] is the capacity and SEER.

The major source of useful data on performance of central air conditioners are the engineering data sheets that some manufacturers provide. Figure 2.1 shows a sample of the data from one manufacturer. Each manufacturer has their own method for presenting the data. For this project, we are most interested in obtaining the sensible heat ratio (SHR) from these tables. The SHR is defined as:

$$\text{SHR} = \frac{\text{Sensible Cooling Capacity (Btu/hr)}}{\text{Total Cooling Capacity (Btu/hr)}} \qquad (2.3)$$

The sensible cooling capacity is the cooling provided by the air conditioner that is only due to a temperature difference between that of the refrigerant and the entering airstream. In most instances, air conditioners will also remove moisture from the incoming airstream. The energy removed (cooling) with the moisture is known as the latent cooling capacity. The higher the SHR, the poorer a unit is at dehumidifying for a given set of conditions.

In Figure 2.1, the sensible capacity ("sens. cap" in the figure) is tabulated as a function of outdoor dry bulb temperature (first column on left), indoor dry bulb temperature(top row), and indoor wet bulb temperature (second column on left). In addition, the total capacity and compressor power are tabulated as functions of outdoor dry bulb and indoor wet bulb temperatures.

PERFORMANCE DATA COOLING

(CAPACITIES ARE NET IN BTUH/1000-INDOOR FAN HEAT DEDUCTED)

BTA036D WITH BXA736D AT 1200 CFM

O.D. D.B.	I.D. W.B.	TOTAL CAP.	SENS. CAP. AT ENTERING D.B. TEMP. 72	74	76	78	80	COMPR. KW	APP.DEW PT.
	59	33.0	26.3	28.4	30.6	32.8	33.7*	3.35	45.7
85	63	35.5	21.9	24.1	26.3	28.5	30.6	3.49	49.7
	67	38.1	17.2	19.4	21.6	23.8	25.9	3.63	53.9
	71	40.8	12.4	14.6	16.7	18.9	21.1	3.78	58.2
	59	32.5	26.1	28.2	30.4	32.6*	33.4*	3.52	45.9
90	63	35.0	21.7	23.9	26.1	28.3	30.4	3.66	49.9
	67	37.5	17.0	19.2	21.4	23.6	25.7	3.81	54.1
	71	40.2	12.2	14.4	16.5	18.7	20.9	3.95	58.4
	59	32.1	25.9	28.0	30.2	32.2*	33.0*	3.70	46.1
95	63	34.4	21.5	23.7	25.9	28.1	30.2	3.83	50.1
	67	37.0	16.8	19.0	21.2	23.3	25.5	3.98	54.3
	71	39.6	12.0	14.2	16.3	18.5	20.7	4.13	58.6
	59	31.3	25.5	27.7	29.9	31.6*	32.4*	3.90	46.4
100	63	33.7	21.2	23.4	25.6	27.7	29.9	4.04	50.4
	67	36.2	16.5	18.7	20.9	23.0	25.2	4.18	54.6
	71	38.7	11.7	13.9	16.0	18.2	20.4	4.33	58.9
	59	30.6	25.2	27.4	29.6	31.0*	31.8*	4.11	46.8
105	63	32.9	20.9	23.1	25.3	27.4	29.6	4.24	50.7
	67	35.3	16.2	18.4	20.5	22.7	24.9	4.38	54.9
	71	37.8	11.4	13.5	15.7	17.9	20.1	4.53	59.3
	59	29.1	24.6	26.8	29.0	29.8*	30.5*	4.54	47.4
115	63	31.3	20.3	22.5	24.6	26.8	29.0	4.65	51.3
	67	33.6	15.6	17.7	19.9	22.1	24.3	4.79	55.6
	71	36.0	10.7	12.9	15.1	17.3	19.5	4.92	59.9

CORRECTION FACTORS - OTHER AIRFLOWS
(multiply or add as indicated)

AIRFLOW	1050	1350
TOTAL CAP.	x0.98	x1.01
SENS. CAP.	x0.94	x1.05
COMPR. KW	x0.99	x1.01
A.D.P.	-1.4	+1.1

VALUES AT ARI RATING CONDITIONS

TOTAL NET CAPACITY = 37000 BTUH
AIRFLOW = 1200 CFM
APP. DEW PT. = 54.3 DEG. F
COMPR. POWER = 3980 WATTS
I.D. FAN POWER = 440 WATTS
O.D. FAN POWER = 280 WATTS
S.E.E.R. = 8.30 BTUH/WATT
E.E.R. = 7.90 BTUH/WATT

NOTE: RATED WITH 25 FEET OF 7/8 SUCT. AND 3/8 LIQUID LINES

* Dry coil condition (Tot. Cap. = Sens. Cap.)
Tot. Cap., Comp. KW and App. Dew Pt. valid only for wet coil. All temperatures in degrees F.

Figure 2.1 - Sample Performance Data from a Manufacturer

The models range in efficiency from a SEER low of 6.4 to a high of 15. This covers a wide range of efficiencies currently on the market. The average SEER of split system central air conditioners sold in 1985 was 8.85[1].

The models range in sensible heat ratios (SHRs) from 0.61 to 0.89 at standard ARI 95 F rating conditions(See Chapter 2). From our discussions with manufacturers, most expressed a desire to keep their units' SHRs below 0.80.

Data Collected

Data from the manufacturers varied slightly. Each manufacturer has its own format for presenting performance and hardware descriptions for their units. Manufacturers such as Trane, Carrier, Lennox and Coleman provide very detailed performance data as a function of outdoor temperature, indoor temperature, and indoor humidity (wet bulb temperature). In contrast, Rheem provides performance data only at one operating condition. Table 3.2 summarizes the data available from the 4 manufacturers providing performance data. Items marked with an asterisk(*) indicate data provided by Rheem.

Table 3.2 - Performance and hardware data collected for this survey.

ITEM	UNITS
Model Number*	
Date of Introduction	year
EER @95F*	btu/(w-hr)
SEER*	btu/(w-hr)
Nominal ARI Capacity @95F*	btu/hr
Indoor Air Flow	cfm
Outdoor Air Flow	cfm
Indoor Coil Area	ft**2
Outdoor Coil Area	ft**2
Sensible Heat Capacity#	btu/hr
Total Capacity#	btu/hr
Compressor Power#	kw

*Only items provided in Rheem literature.
#Data was tabulated for combinations of 95 F outdoor temperature, indoor dry bulb temperatures of 72, 76, and 80 F, and indoor wet bulb temperatures of 63, 67, and 71 F.

References

1. "Directory of Certified Unitary Air Conditioners, Air Source Heat Pumps, and Sound-Rated Outdoor Unitary Equipment", Semi-Annual Issues, Air Conditioning and Refrigeration Institute, Arlington, VA.

2. "Test Procedure for Central Air Conditioners, Including Heat Pumps", Federal Register, Dec. 27, 1979, pp. 76700-76723.

CHAPTER 4
RESULTS

The dehumidification performance of central air conditioners was regressed with other performance and hardware characteristics to assess what variables may influence dehumidification performance. Dehumidification performance was measured in SHR. The major performance variables and hardware characteristics included: SEER, Net Cooling Capacity/Outdoor Coil Face Area, Net Cooling Capacity/Indoor Coil Face Area, and Indoor Coil Face Velocity. Each of these is discussed below.

SEER versus SHR

Figure 4.1 shows the SEER plotted as a function of SHR for air conditioners from all manufacturers. The SHR is for the standard ARI rating conditions (95 F outdoor, 80 F indoor dry bulb, and 67 F indoor wet bulb temperatures). There is not a strong correlation between efficiency (as measured in SEER) and SHR. Both linear and quadratic correlations were attempted for the data. The linear correlation was in the form:

$$SEER = A*SHR + B \tag{4.1}$$

The quadratic correlation was in the form:

$$SEER = C*SHR^{2} + D*SHR + E \tag{4.2}$$

where A,B,C,D and E are constants from the regression. Table 4.1 presents the constants.

Table 4.1 - Regression constants for the curve fit of SEER to SHR

Regression	Constant	Value
Linear	A	13.95
	B	- 0.90
Quadratic	C	127.38
	D	-181.08
	E	73.52

The slope of the linear fit is positive, indicating that an increase in SEER produces an increase in SHR (or vice-versa). The slope would indicate that for every increase in SEER of 1.0, you should expect an increase in SHR of 0.07. However, the R-squared for both the linear and quadratic fits are low. The linear fit only has an R-squared of 0.21, while the quadratic only has an R-squared of 0.29. Thus, the linear fit is only able to explain 21% of the variation in the data, which means 79% of the variation is due to other variables.

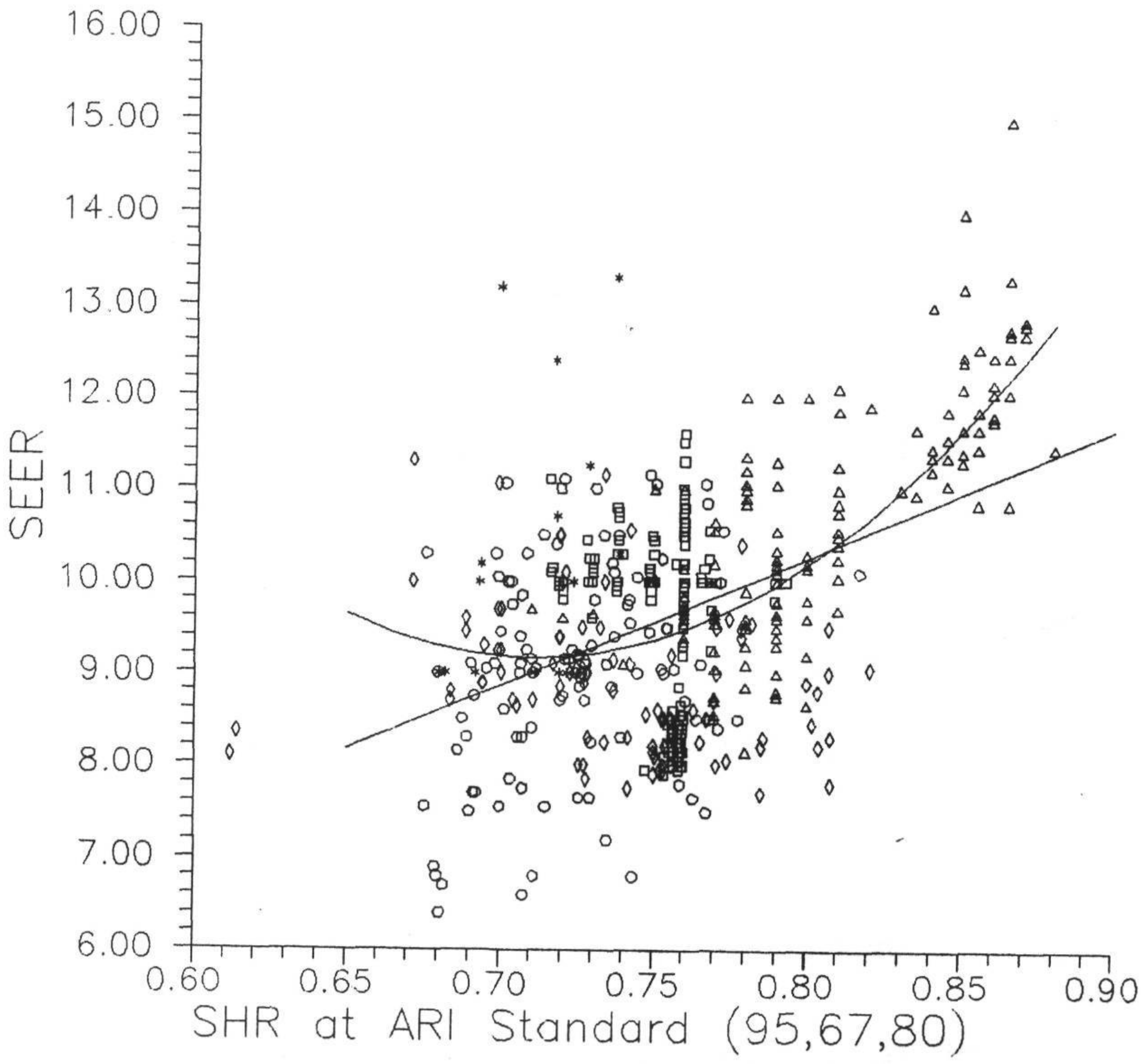

Figure 4.1 - SEER vs SHR for the Five Manufacturers Data

The wide variation in data could have several explanations. First, a manufacturer can choose several design routes to improving the efficiency of their systems. For example, a manufacturer could improve the efficiency of a system by increasing/improving the heat transfer surfaces, installing a more efficient compressor, changing airflow rates, etc. Each improvement has a different effect on the SHR. Using larger heat transfer surfaces on the the evaporator allow the manufacturer to increase refrigerant evaporating temperatures and maintain the same overall capacity. However, increasing the evaporator temperature will decrease the SHR. In contrast, increasing the efficiency by installing a higher efficiency compressor should produce little change in the SHR. One manufacturer may favor one approach over the other due to a design philosophy within the company.

Another possible variation in the data is due to the fact that the units available in 1986 were introduced over several years. A particular design philosophy that was incorporated in models introduced in 1983 may be different from those introduced in 1986.

To test for differences in design strategy, we decided to correlate manufacturers' data separately. Figures 4.2 through 4.6 show plots of SHR vs. SEER for Carrier, Trane, Lennox, Rheem, and Coleman, respectively. Linear regressions were made to the data and are presented in Table 4.2. The most surprising result is the overall lack of

Table 4.2 - Regression results for the five manufacturers.

Manufacturer	Regression Coefficient A	Regression Coefficient B	R-Squared
Carrier	-13.56	19.75	0.05
Trane	11.94	30.41	0.09
Lennox	25.36	- 9.74	0.57
Rheem	- 6.47	13.90	0.08
Coleman	19.54	- 3.66	0.10

relationship between SEER and SHR. With the exception of Lennox, all manufacturers had R-squared values of 0.10 or less. One can only conclude that SHR is only weakly related to SEER for these four manufacturers. Lennox's units produced the highest correlation with an R-squared of 0.57. Lennox also had the largest number of high (above 10) SEER air conditioners. The SHRs for most Lennox's high SEER units were above 0.80. This contrasts with Coleman's high SEER units, which had SHRs below 0.80. This would indicate that Coleman is able to build high efficiency air conditioners that maintain reasonable dehumidification characteristics.

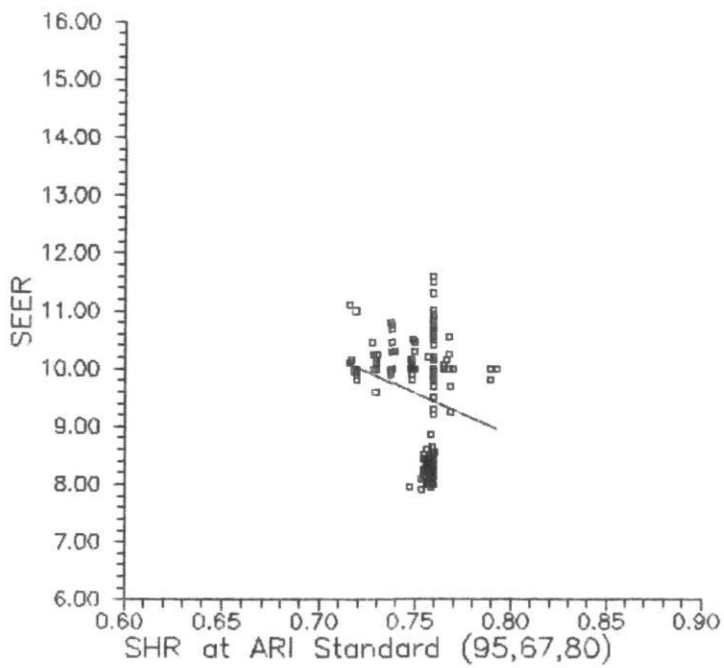

Figure 4.2 - SEER vs SHR for Carrier Data

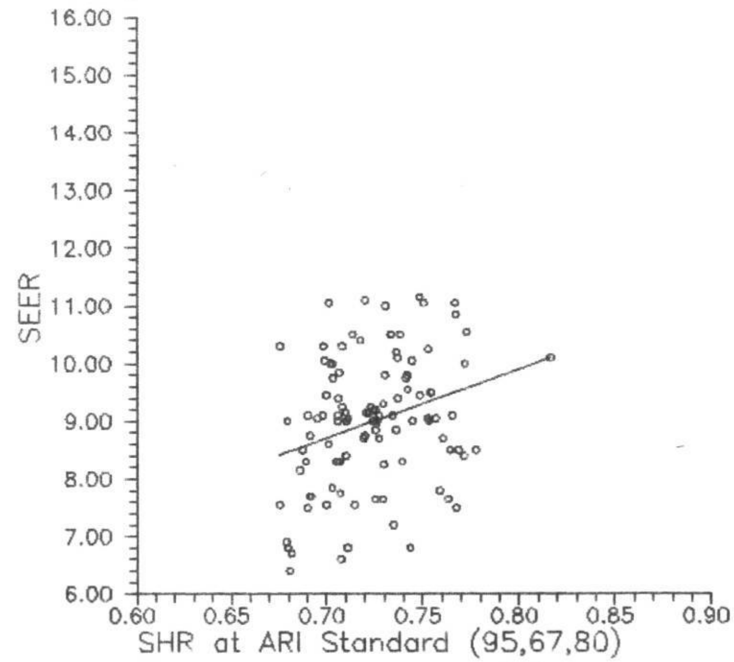

Figure 4.3 - SEER vs SHR for Trane Data

Recall that the ARI rating conditions (Test A) are run with an outdoor temperature of 95 F, indoor temperature of 80 F and indoor wet bulb temperture of 67 F. For the unit shown in Figure 2.1, the sensible capacity would be 25.5 x 10**3 Btu/hr and the total capacity would be 37.0 x 10**3 Btu/hr. This yields an SHR of:

$$\mathrm{SHR} = \frac{25.5 \times 10^{3}\ \mathrm{Btu/hr}}{37.0 \times 10^{3}\ \mathrm{Btu/hr}} = 0.69 \qquad (2.4)$$

The SHR will vary with the inlet air conditions to the indoor coil. Generally, the greater the moisture is in the entering air, the smaller the SHR and the greater the dehumidification.

References

1. "Directory of Certified Unitary Air Conditioners, Air Source Heat Pumps, and Sound-Rated Outdoor Unitary Equipment", Semi-Annual Issues, Air Conditioning and Refrigeration Institute, Arlington, VA.

2. "Test Procedure for Central Air Conditioners, Including Heat Pumps", Federal Register, Dec. 27, 1979, pp. 76700-76723.

CHAPTER 3
SURVEY DESCRIPTION

This chapter describes the survey done and the data collected from the survey of manufacturers.

Survey

The survey was made of five manufacturers (Lennox, Carrier, Trane, Rheem, and Coleman) on their models of air conditioners available on the market in 1986. This choice of manufacturers includes the two largest manufacturers (Trane and Carrier) of residential air conditioners and three medium sized manufacturers (Lennox, Rheem and Coleman). Approximately 600 different central air conditioner models* were included in the survey. The models were limited to split systems, which constitute over 85% of total shipments of central systems[1]. The models represent a large cross section of split system central air conditioners. Table 3.1 lists the total number of units included in the survey from each manufacturer.

Table 3.1 - Air conditioner models included from each manufacturer.

Manufacturer	Number of Units Included
Carrier	212
Coleman	17
Trane	138
Rheem	110
Lennox	108
TOTAL	585

The models range in capacities from 14000 Btu/hr to 58000 Btu/hr. The smaller models would be more typical of units installed in small apartments while the larger units would be typical of installations in larger homes or small commercial buildings.

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*A model constitutes a specific evaporator/condenser combination. Typically, a manufacturer has several evaporators that can be matched to a condenser. Thus, two of the models included in the survey may have the same condenser but different evaporators.

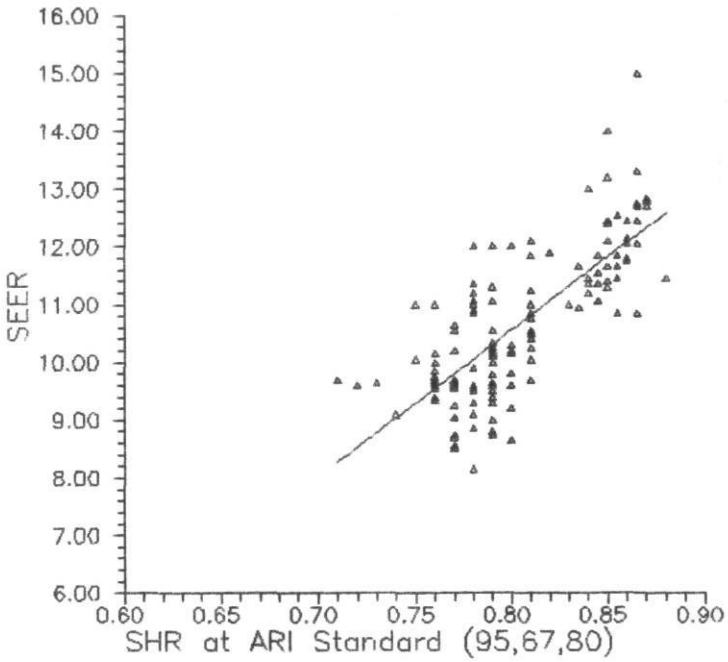

Figure 4.4 - SEER vs SHR for the Lennox Data

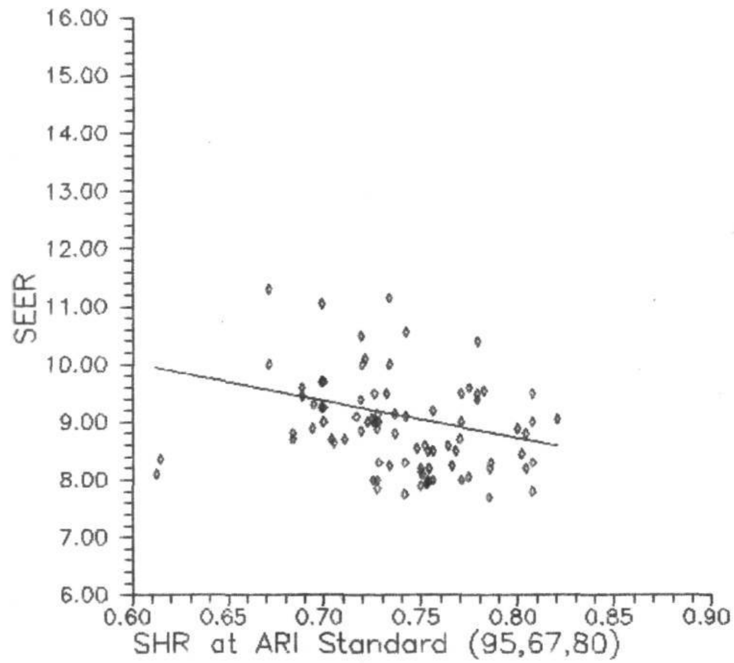

Figure 4.5 - SEER vs SHR for the Rheem Data

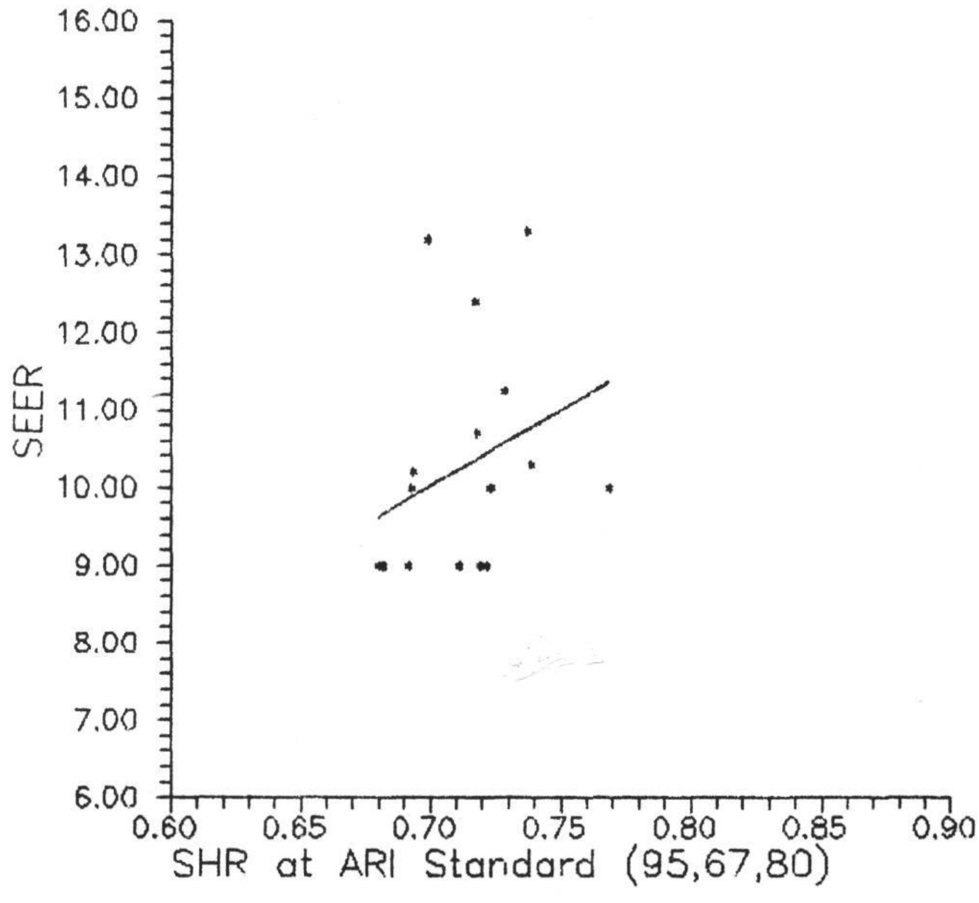

Figure 4.6 - SEER vs SHR for the Coleman Data

SHR versus Outdoor Coil Size

Since there was no significant correlation between SHR and SEER, other possible variables were considered that may help in predicting SHR trends. One of these variables was the outdoor heat exchanger (or coil) size.

As mentioned earlier, one method that manufacturers can increase the efficiency of their systems is through increasing the area of heat transfer surfaces on coils. This can be accomplished through increasing the face (or frontal) area, number of tube rows, and fin density (fins per inch). Many manufacturers already have high fin densities (greater than 17 fins/inch) and normally do not use more than two tube rows on their outdoor coils. Increasing the outdoor face area will result in a unit with larger external dimensions.

Figures 4.7 through 4.9 show plots of SEER as a function of the ratio, net cooling capacity (in Btu/hr) divided by the outdoor coil face area (in ft**2), for Carrier, Trane and Lennox, respectively. Because manufacturers use larger coils to increase cooling capacity, the cooling capacity was divided by the coil face area to get a normalized value of the face area. In the two figures, increasing face area is represented by a shift to the left on the x-axis. In Figures 4.7 through 4.9, the SEER increased with increasing values of the outdoor coil face area. The R-squared for the regressions on the 3 figures was 0.23 for Carrier, 0.67 for Trane and 0.34 for Lennox.

While increasing outdoor coil size brings improvements in the SEER, the data indicate that it has very little effect on the SHR. Figures 4.10 through 4.12 show plots of SHR versus the ratio of net cooling capacity/outdoor coil face area for Carrier, Trane, and Lennox, respectively. Table 4.3 shows the R-Squared for each correlation. Lennox was the only manufacturer whose units showed

Table 4.3 - R-Squared Values for regressing SHR to cooling capacity/outdoor coil face area.

Manufacturer	R-Squared
Carrier	0.10
Trane	0.03
Lennox	0.26

even a modest correlation between the SHR and outdoor face area. The Trane data yielded almost no correlation (R-Squared of 0.03).

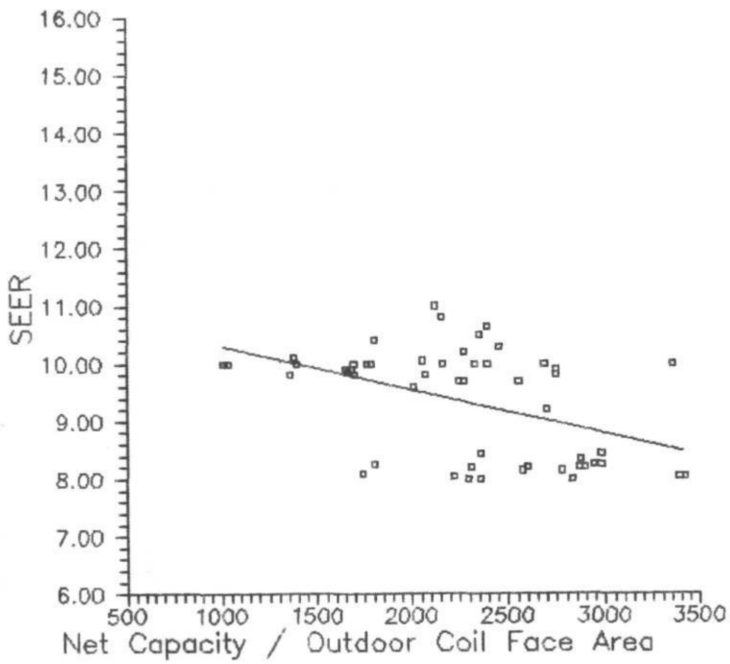

Figure 4.7 - SEER vs Net Capacity/Outdoor Coil Face Area for the Carrier Data

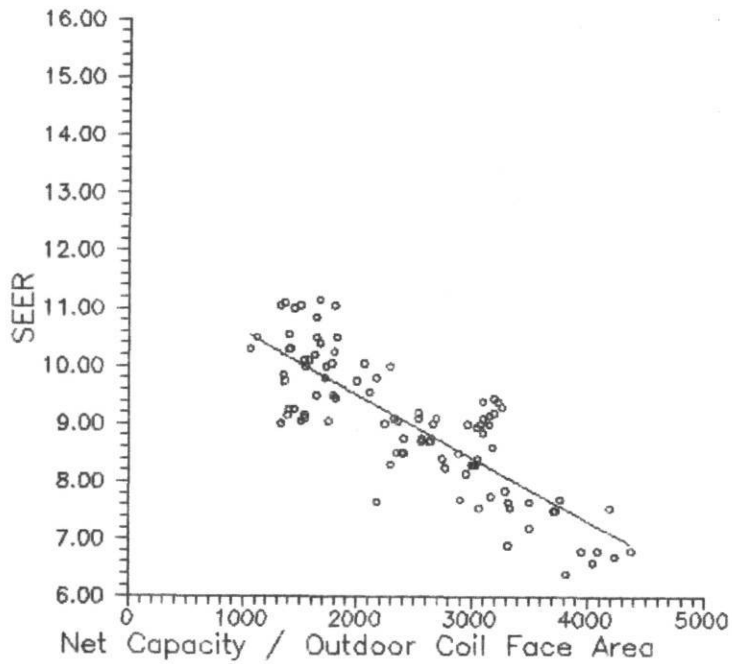

Figure 4.8 - SEER vs Net Capacity/Outdoor Coil Face Area for the Trane Data

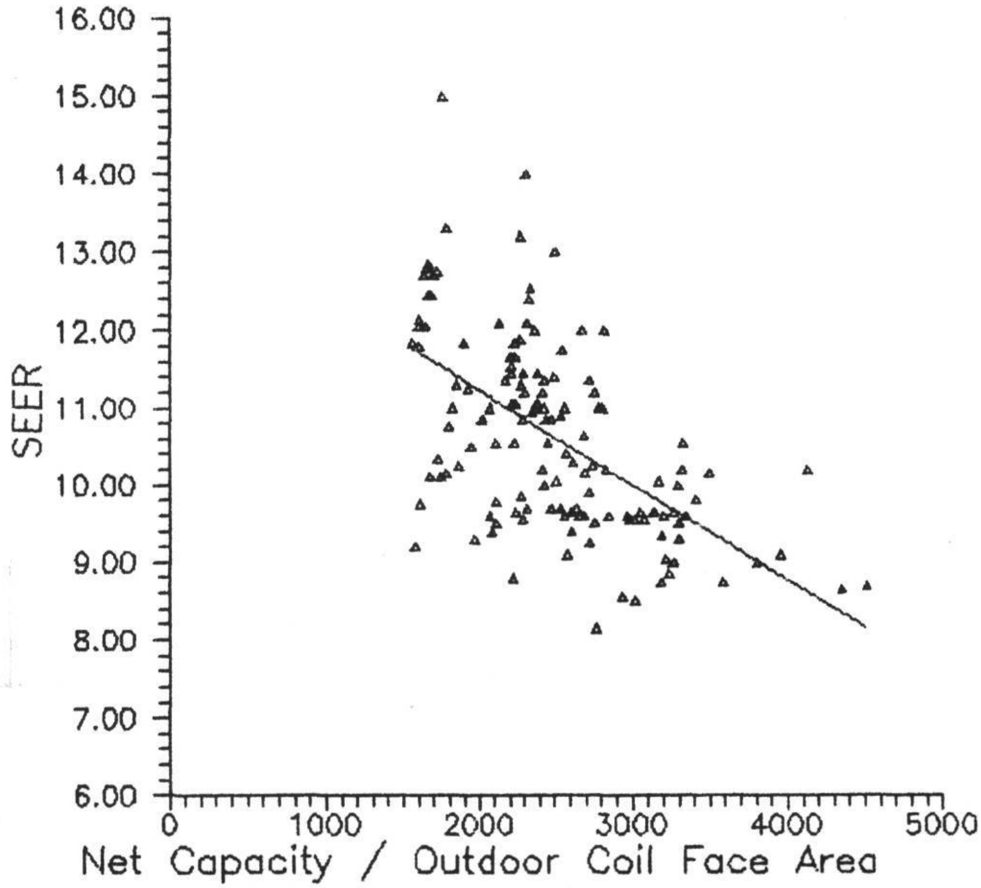

Figure 4.9 - SEER vs Net Capacity/Outdoor Coil Face Area for the Lennox Data

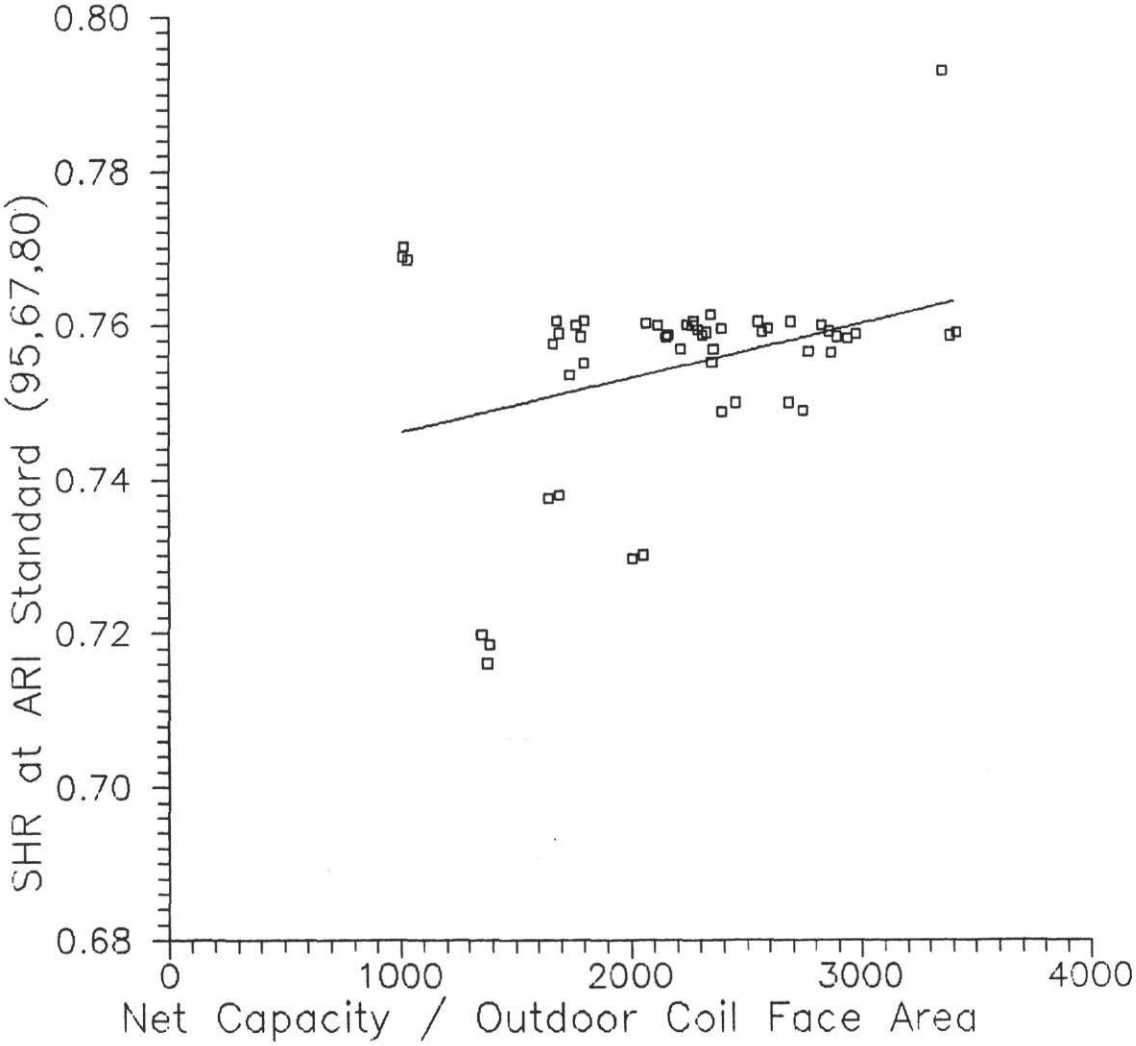

Figure 4.10 - SHR versus Net Capacity/Outdoor Coil Face Area for the Carrier Data

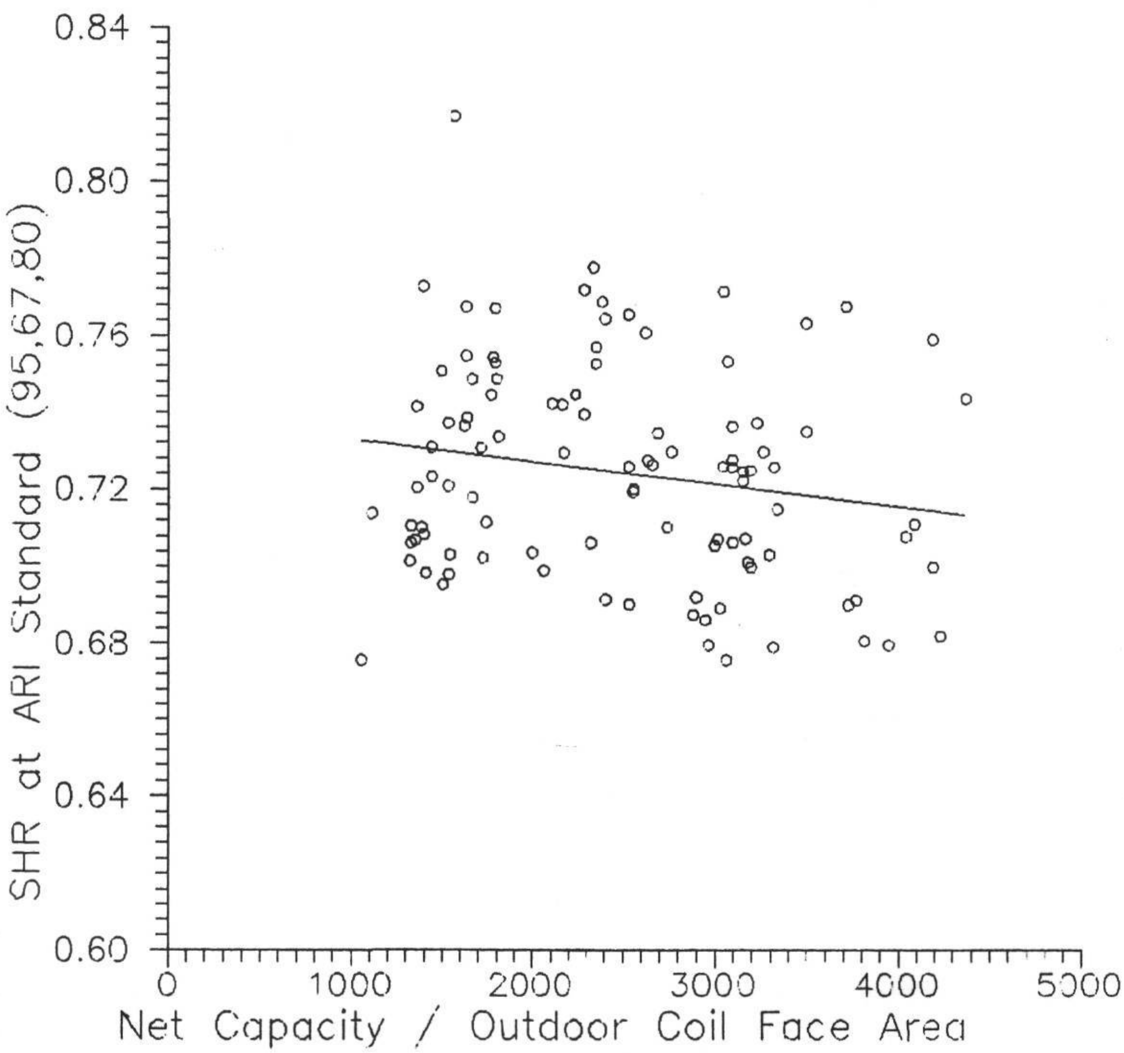

Figure 4.11 - SHR versus Net Capacity/Outdoor Coil Face Area for the Trane Data

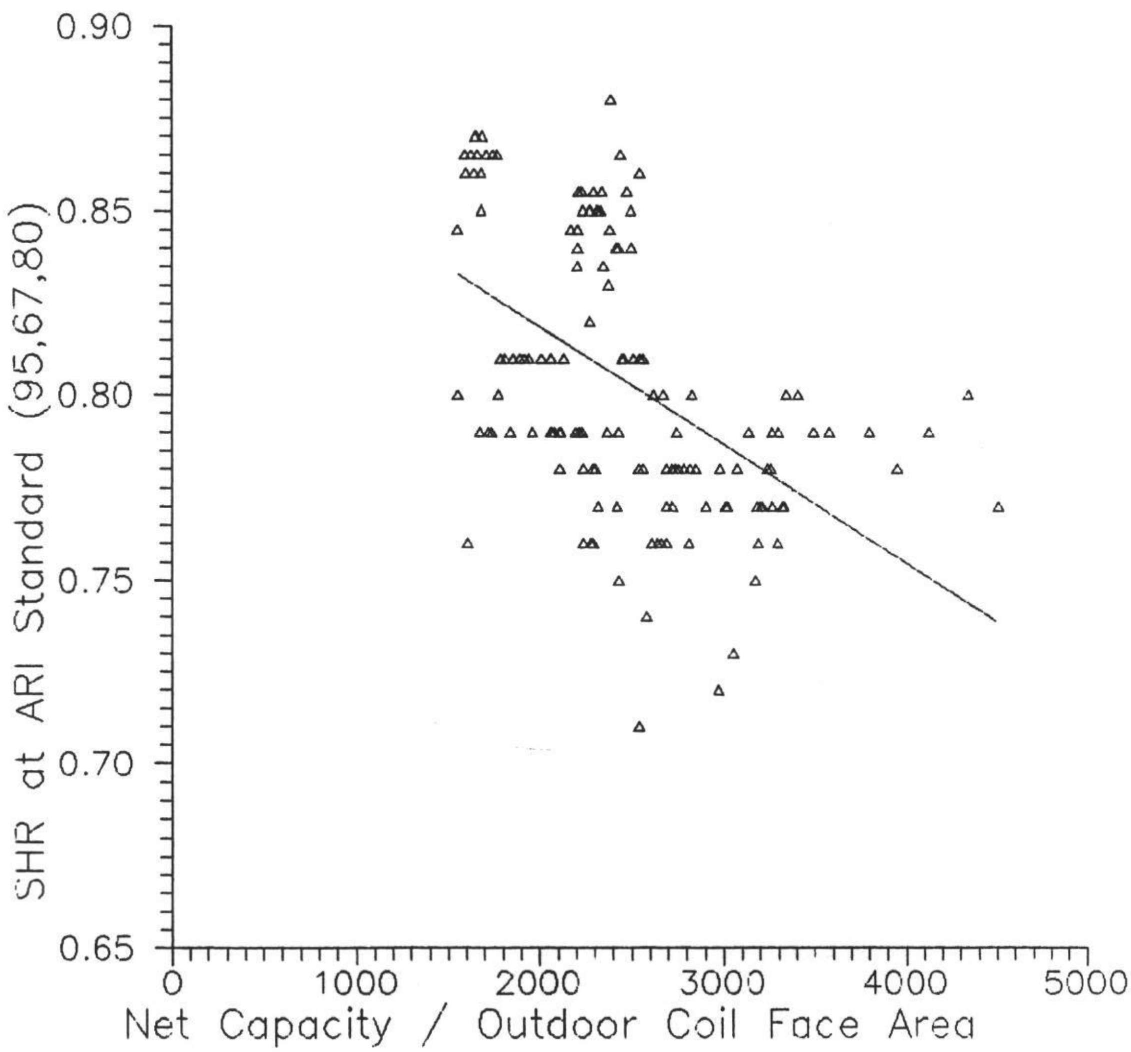

Figure 4.12 - SHR versus Net Capacity/Outdoor Coil Face Area for the Lennox Data

SHR versus Indoor Coil Size

Another variable that could affect the SHR is the indoor coil size. As with the outdoor coil, manufacturers have increased indoor coil sizes over the past few years to increase SEERs. However, manufacturers have more severe size restrictions on the indoor coil than the outdoor coil. For central air conditioners, the indoor coil must fit into the ductwork typically associated with the furnace. Another restriction on the indoor coil is the fin spacing. Typically, manufacturers like to restrict the indoor coil fin density to 12 to 14 fins per inch to allow for enough space for the condensate to run off the coil.

As with the outdoor coil, the SEER was plotted against the ratio of the net capacity divided by the indoor coil face area. Figures 4.13 through 4.15 shows these plots for Carrier, Trane, and Lennox units, respectively. All figures show a general trend of increasing SEER with increasing indoor coil face area. However, the data from all manufacturers show significant scatter. R-Squareds ranged from 0.05 for Lennox to 0.08 for Carrier.

Plots of the indoor coil face area and the SHR shows considerable scatter (Figures 4.16 through 4.18). Table 4.4 gives the R-Squared values for each manufacturer. There appears to be little or no

Table 4.4 - R-Squared values for regressing SHR to cooling capacity/indoor coil face area.

Manufacturer	R-Squared
Carrier	0.03
Trane	0.12
Lennox	0.02

correlation between the indoor coil size and the SHR. The highest R-Squared was 0.12 for the Trane units, which means that 88% of the scatter in their SHR data is not explained by the indoor coil size.

SHR versus Indoor Coil Face Velocity

The last variable that was examined to see if it could explain the differences in SHR was the indoor coil face velocity. One way to decrease the SHR (improve dehumidification) is to lower the face velocity on the indoor coil. This allows the moist air to remain in contact with the cold heat transfer surfaces on the coil longer which improves moisture removal from the air.

Manufacturers do not directly provide the indoor face velocity. It must be calculated from the face area and the air flow rate. The face velocity, FV, is given as:

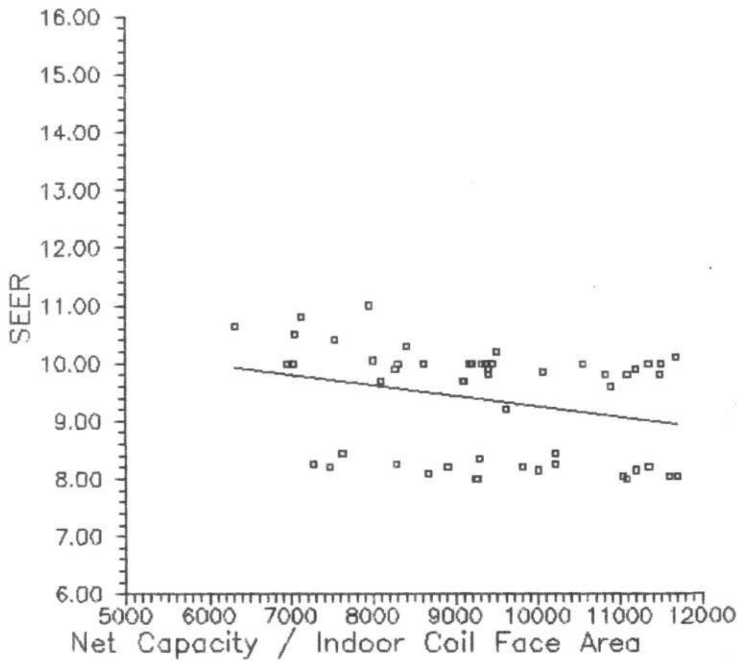

Figure 4.13 - SEER vs Net Capacity/Indoor Coil Face Area for the Carrier Data

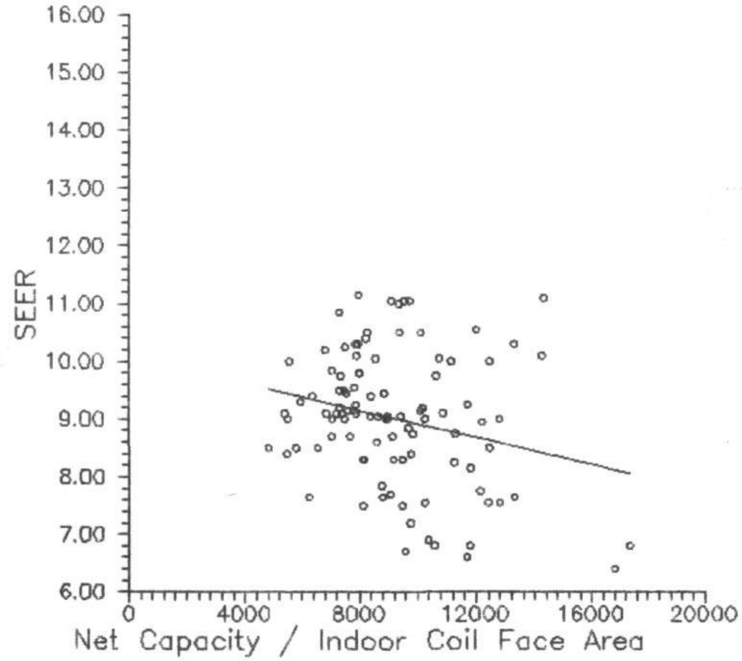

Figure 4.14 - SEER vs Net Capacity/Indoor Coil Face Area for the Trane Data

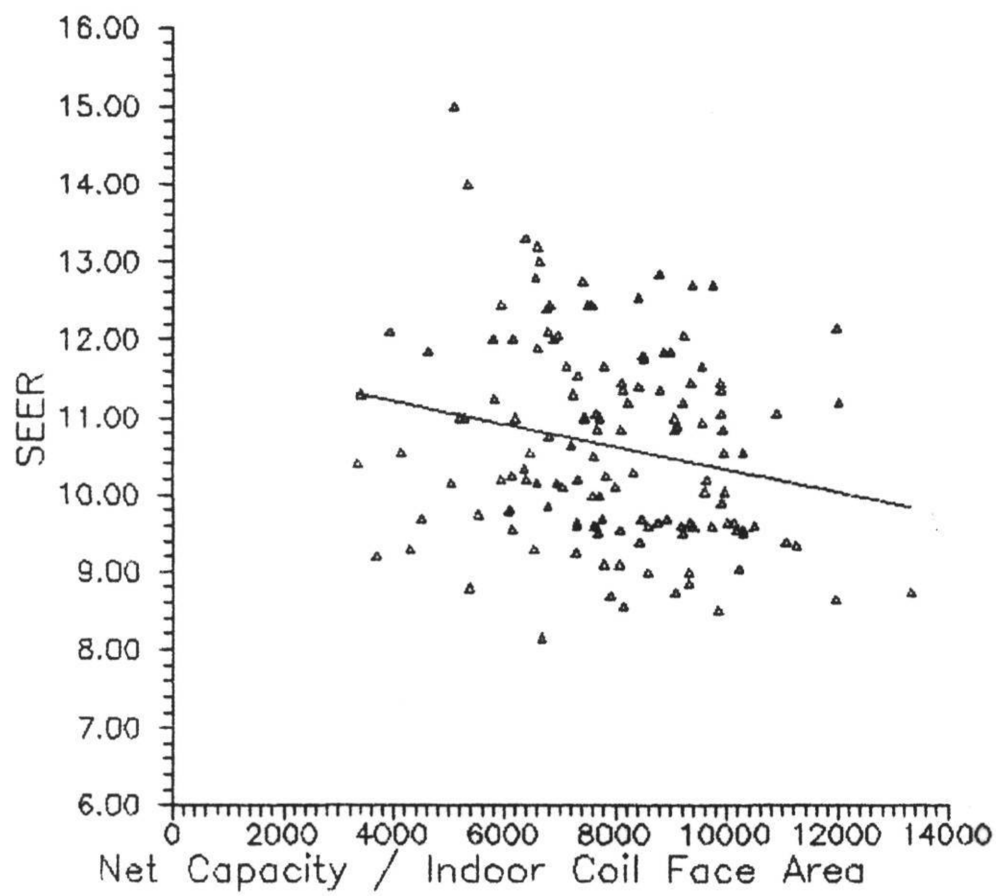

Figure 4.15 - SEER vs Net Capacity/Indoor Coil Face Area
for the Lennox Data

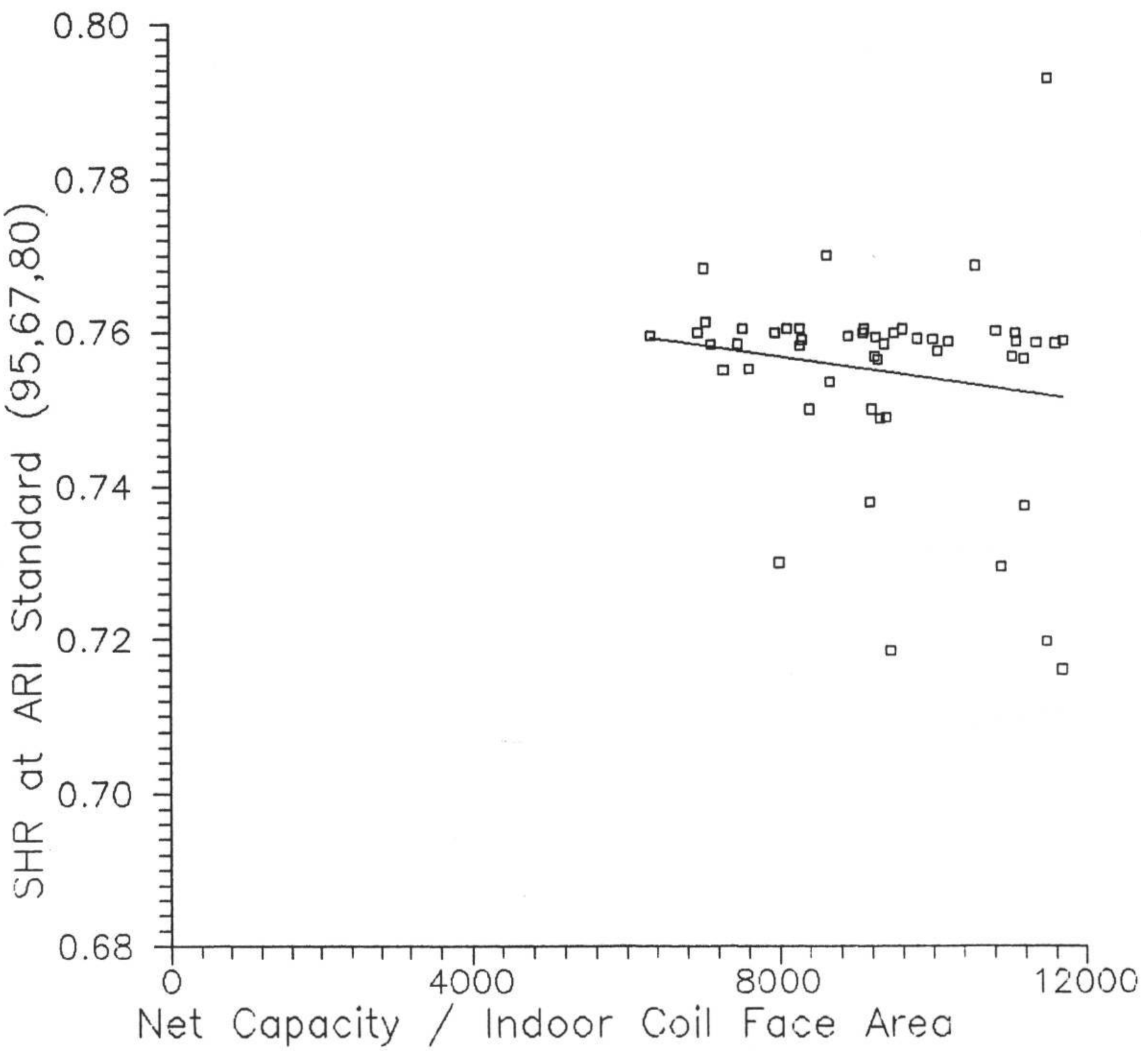

Figure 4.16 - SHR vs Net Capacity/Indoor Coil Face Area for the Carrier Data

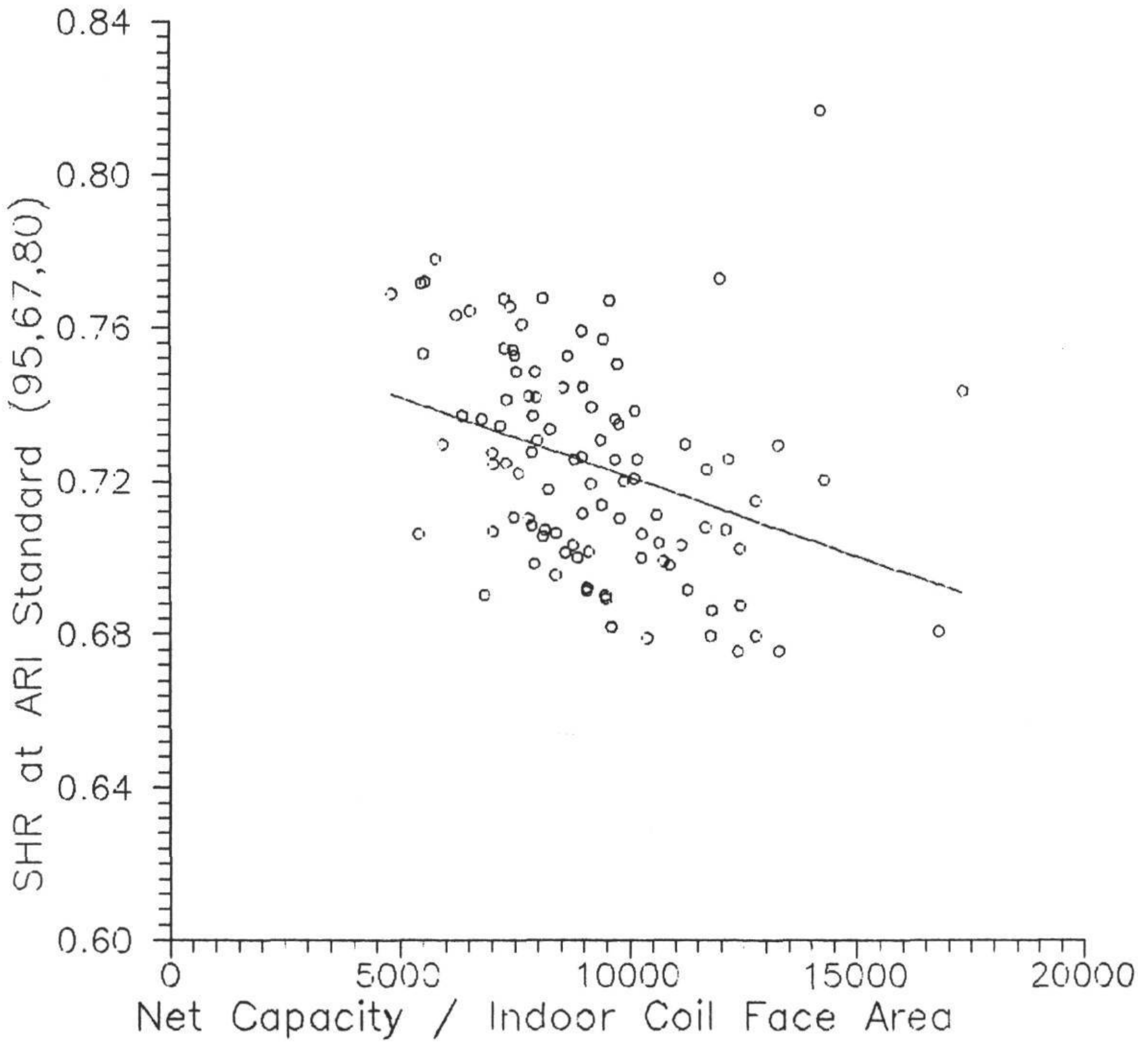

Figure 4.17 - SHR vs Net Capacity/Indoor Coil Face
Area for the Trane Data

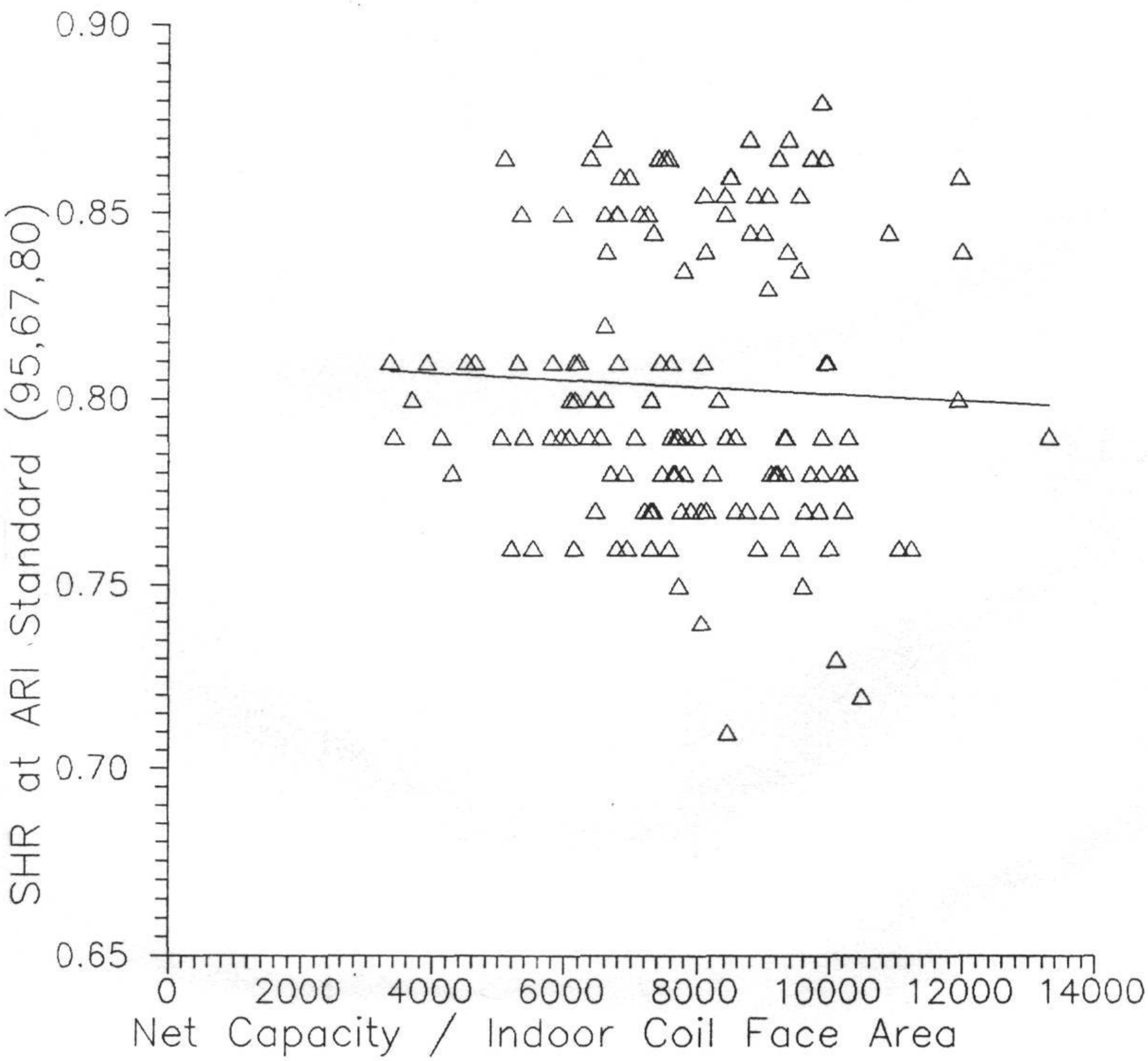

Figure 4.18 - SHR vs Net Capacity/Indoor Coil Face Area for the Lennox Data

$$FV = \frac{\text{Indoor Air Flow Rate (CFM)}}{\text{Indoor Coil Face Area (ft**2)}} \qquad (4.1)$$

Only Carrier, Trane, and Lennox provided enough data to estimate the indoor coil face velocity.

Figures 4.19 through 4.21 show the plots of SHR as a function of the indoor face velocity. As with the other variables, there is much scatter in the data. Table 4.5 gives the R-Squared values of the correlations.

Table 4.5 - R-Squared values for regressing SHR to indoor face velocity.

Manufacturer	R-Squared
Carrier	0.03
Trane	0.01
Lennox	0.03

Summary

While there may be a relationship between SHR and SEER or some of the other hardware variables examined in this chapter, the data from the five manufacturers do not support a very strong relationship. For some of the variables, such as indoor face velocity or indoor coil size, the R-Squared values are so low as to suggest there is no relationship between SHR and those variables.

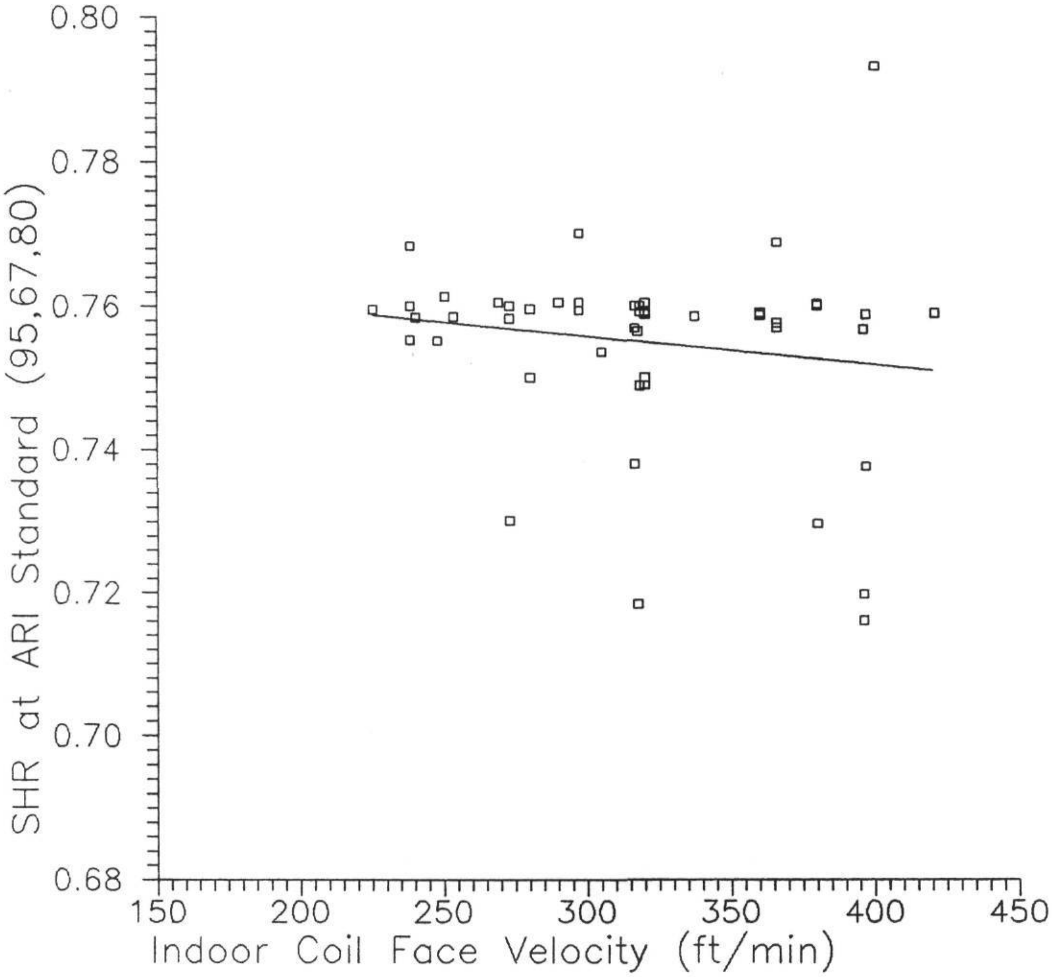

Figure 4.19 - SHR vs Indoor Coil Face Velocity for the Carrier Data

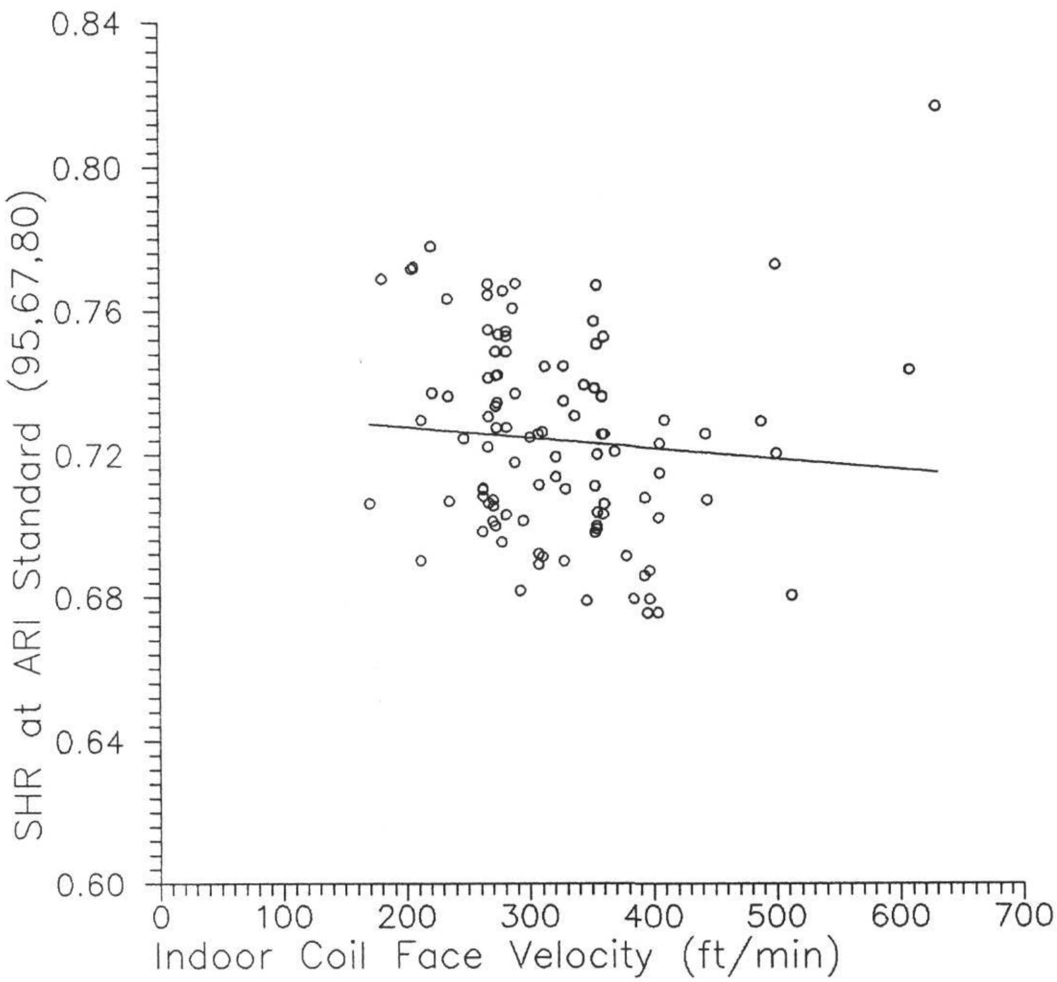

Figure 4.20 - SHR vs Indoor Coil Face Velocity
for the Trane Data

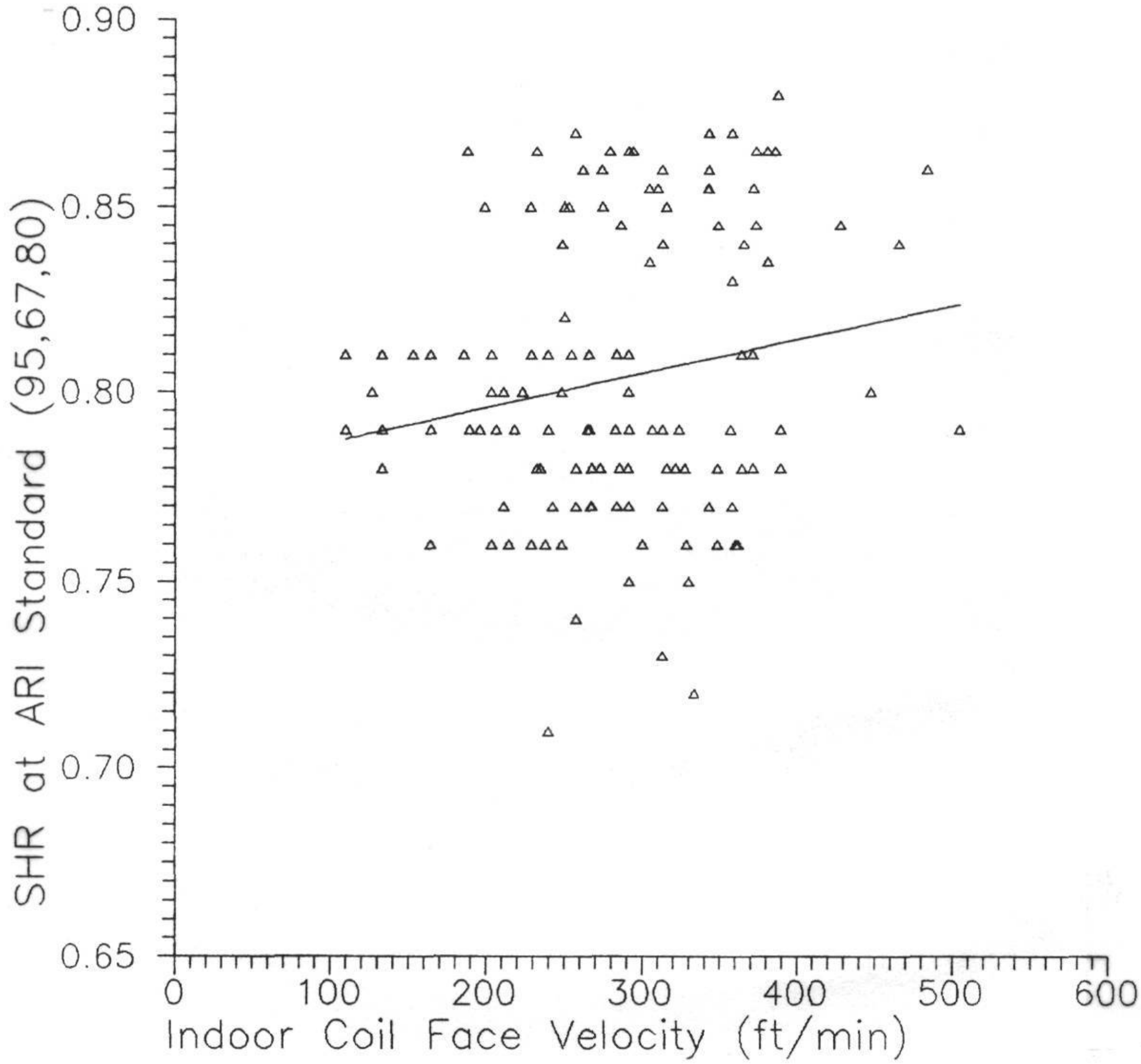

Figure 4.21 - SHR vs Indoor Coil Face Velocity
for the Lennox Data

CHAPTER 5
CONCLUSIONS AND RECOMMENDATIONS

Air conditioning manufacturers have made significant improvements in the efficiency of split system central air conditioners during the past ten years. Average EERs/SEERs have increased by 23%, from 7.16 to 8.84, during this period. With the added incentive of minimum efficiency standards to be enforced in 1992, manufacturers will continue improving the efficiency of central air conditioners[1].

The dramatic improvements in average SEERs over the past ten years have been overshadowed by the improvements at the higher end of the efficiency spectrum. In 1976, there were only five central air conditioner models with EERs greater than 9.2 out of a possible 1800 offered by the air conditioning industry[2]. In written testimony to the California Energy Commission in 1976, Mr. George Hudelson, Vice President for Engineering of Carrier Corporation stated that the units with EERs greater than 9.2 were "simply the result of fortunate circumstance...Occasionally, certain components come together in a design where everything peaks at exactly the optimum point." Since 1976, manufacturers have learned how to design high efficiency air conditioners. Of the five manufacturers surveyed in this report, all of them offered units with SEERs above 11 and two (Lennox and Coleman) offer unit with SEERs above 13. In addition, Carrier and Trane have announced variable speed units with SEERs above 15.

Improvements in efficiency at the high end have not always resulted in better dehumidification characteristics. Lennox's high efficiency units (typically two-speed) have SHRs greater than 0.85, indicating that they probably may not adequately dehumidify for Houston applications. However, Coleman has produced units with both high SEERs and low SHRs, indicating that it is possible to have an air conditioner which can both dehumidify and save energy.

The data in the survey indicated that there was only a weak (at best) relationship between the SHR and any of the other variables examined (SEER, outdoor coil area, indoor coil area, and indoor face velocity). Most correlations of SHR with these variables showed very small R-Squared values.

It would appear that more detailed hardware/performance data would be necessary to adequately predict what hardware configuration might affect the SHR. This hardware data might include compressor model numbers, compressor performance maps, evaporating temperatures, etc. Much of this type of data would have to come directly from the manufacturer who may be reluctant to provide it. Even with this expanded data, it does not guarantee that good correlations could be developed. Thus, we do not recommend an extension of the current survey.

The published data that manufacturers provide is steady-state. The actual performance of a central air conditioner will be determined as the unit cycles on and off in a residence under transient latent and sensible loads. Thus, even if the manufacturer's SHR data could be correlated to other performance/hardware characteristics, it still would not indicate whether air conditioners adequately dehumidify in Houston.

To assess whether the units dehumidify well in Houston, we recommend first modeling typical residences in Houston with air conditioners having a wide range of SHR. This would indicate what level of SHR that is adequate in a Houston climate. Ideally, this work should be coupled with an experimental program to verify the model and the manufacturer's performance data.

References

1. "A Bill to Amend the Energy Policy and Conservation Act with Respect to Energy Conservation Standards for Appliances", 100th Congress, Report No. 100-6, Bill S.83.

2. "Directory of Certified Unitary Air Conditioners, Air Source Heat Pumps, and Sound-Rated Outdoor Unitary Equipment", Jan. 1 - June 30, 1976, Air Conditioning and Refrigeration Institute, Arlington, VA.

3. Letter to the California State Energy Resources Conservation and Development Commission, State of California, from George D. Hudelson, Vice President - Engineering, Carrier Corporation, Dated August 10, 1976.

APPENDIX

UNITARY AIR-CONDITIONING AND AIR SOURCE HEAT PUMP EQUIPMENT

Section 1. Purpose

1.1. *Purpose.* The purpose of this standard is to establish, for unitary equipment: definitions and classification; requirements for testing and rating; specifications, literature and advertising requirements; performance requirements; safety recommendations and conformance conditions.

1.1.1. This standard is intended for the guidance of the industry, including manufacturers, distributors, installers, contractors, consumers and all levels of government.

1.2 *Review and Amendment.* This standard is subject to review and amendment as the art of the industry advances.

Section 2. Scope

2.1 *Scope.* This standard applies to factory-made residential, commercial, and industrial equipment defined in 3.1 and 3.2.

2.1.1 *Energy Source.* This standard applies only to electrically-driven, mechanical compression type systems.

2.2 *Exclusions.* This standard does not apply to the rating and testing of individual assemblies, such as condensing units or coils, for separate use.

2.2.1 This standard does not apply to heat operated air-conditioning/heat pump equipment, or packaged terminal air-conditioners/heat pumps, or to room air-conditioners/heat pumps.

2.2.2 This standard does not apply to unitary air-conditioners as defined in ARI Standard 360, with capacities 135,000 Btuh (40 kW) or greater.

2.2.3 This standard does not apply to unitary heat pumps as defined in ARI Standard 340, with cooling capacities of 135,000 Btuh (40 kW) or greater, nor to water source heat pumps as defined in ARI Standard 320, nor to ground water source heat pumps as defined in ARI Standard 325.

2.2.4 This standard does not include water heating heat pumps.

2.2.5 This standard does not apply to units equipped with desuperheater/water heating device.

Section 3. Definitions

3.1 *Air-Source Unitary Heat Pump.* An air-source unitary heat pump consists of one or more factory-made assemblies which normally includes an indoor conditioning coil(s), compressor(s) and outdoor coil(s), including means to provide a heating function. When such equipment is provided in more than one assembly, the separated assemblies shall be designed to be used together, and the requirements of rating outlined in the standard are based upon the use of matched assemblies.

3.1.1 *Functions.* Unitary heat pumps shall provide the function of air heating with controlled temperature, and may include the functions of air-cooling, air-circulating, air-cleaning, dehumidifying or humidifying.

3.2 *Unitary Air-Conditioner.* An unitary air-conditioner consists of one or more factory-made assemblies which normally include an evaporator or cooling coil(s), compressor(s) and condenser(s). Where such equipment is provided in more than one assembly, the separated assemblies are to be designed to be used together, and the requirements of rating outlined in this standard are based upon the use of matched assemblies.

3.2.1 *Functions.* The functions of unitary air-conditioners either alone or in combination with a heating plant, are to provide air-circulation, air-cleaning, cooling with controlled temperature and dehumidification, and may optionally include the function of heating and possible humidifying.

3.3 *Published Rating.* A published rating is a statement of the assigned values of those performance characteristics, under stated rating conditions, by which a unit may be chosen to fit its application. These values apply to all units of like nominal capacity and type (identification) produced by the same manufacturer. As used herein, the term "published rating" includes the rating of all performance characteristics shown on the equipment or published in specifications, advertising, or other literature controlled by the manufacturer, at stated rating conditions.

3.4 *Rating Conditions.* Rating conditions are any set of operating conditions under which a single level of performance results, and which causes only that level of performance to occur.

3.5 *Energy Efficiency Ratio (EER).* EER is a ratio calculated by dividing the net cooling capacity in Btu/h by the power input in watts (W) at any given set of rating conditions, expressed in Btu/Wh.

3.6 *Seasonal Energy Efficiency Ratio (SEER).* Seasonal Energy Efficiency Ratio means the total cooling of a central air-conditioner in Btu's during its normal usage period for cooling (not to exceed 12 months) divided by the total electric energy input in watt-hours during the same period as determined in Appendices A and/or B, and ASHRAE Standard 116 *Methods of Testing For Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps* (American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.).

3.7 *Coefficient of Performance (COP) Heating.* COP is a ratio calculated by dividing the total heat capacity provided by the refrigeration system including circulating fan heat, but excluding supplementary resistance heat, by the total electrical input in consistent units at any given set of rating conditions.

3.8 *Heating Seasonal Performance Factor (HSPF).* HSPF is the total heating output of a heat pump, including supplementary electric heat in Btu necessary to achieve building heating requirements during its normal annual usage period for heating divided by the total electric power in watt-hours (Wh) during the same period, as determined in Appendices A and/or B, and ASHRAE Standard 116.

3.9 *Degradation Coefficient (C_D).* The C_D is the measure of the efficiency loss due to the cycling of the units as determined in Appendices A and/or B, and ASHRAE Standard 116.

3.10 *"Shall", "Should", "Recommended", or "It Is Recommended".* "Shall", "should", "recommended" or "it is recommended" shall be interpreted as follows:

3.10.1 *Shall.* Where "shall" or "shall not" is used for a provision specified, that provision is mandatory if compliance with the standard is claimed.

3.10.2 *Should, Recommended, or It is Recommended.* "Should", "recommended", or "It is recommended" is used to indicate provisions which are not mandatory but which are desirable as good practice.

Section 4. Classification

4.1 *Classification.* Normally, unitary equipment within the scope of this standard may be classified as shown in Table 1 and Table 2.

Table 1 — Classification of Unitary Air-Conditioners

Types of Unitary Air-Conditioners			
Designation	ARI Type	Heat Rejection	Arrangement
Single Package	SP-A SP-E SP-W	Air Evap Cond Water	FAN | COMP EVAP | COND
Refrigeration Chassis	RCH-A RCH-E RCH-W	Air Evap Cond Water	COMP EVAP | COND
Year-Round Single Package	SPY-A SPY-E SPY-W	Air Evap Cond Water	FAN HEAT | COMP EVAP | COND
Remote Condenser	RC-A RC-E RC-W	Air Evap Cond Water	FAN EVAP COND COMP
Year-Round Remote Condenser	RCY-A RCY-E RCY-W	Air Evap Cond Water	FAN EVAP COND HEAT COMP
Condensing Unit Coil Alone	RCU-A-C RCU-E-C RCU-W-C	Air Evap Cond Water	EVAP COND COMP
Condensing Unit Coil and Blower	RCU-A-CB RCU-E-CB RCU-W-CB	Air Evap Cond Water	FAN COND EVAP COMP
Year-Round Condensing Unit and Blower	RCUY-A-CB RCUY-E-CB RCUY-W-CB	Air Evap Cond Water	FAN EVAP COND HEAT COMP

Extracted by permission from Table 1, Chapter 42, 1979 ASHRAE Handbook.

A suffix of "-0" following any of the above classifications indicates equipment not intended for use with field-installed duct systems (See 5.1.3.3b)

Table 2 — Classification of Unitary Heat Pumps

Types of Unitary Air-Source Heat Pumps			
	† ARI Type		
Designation	Heating and Cooling	Heating Only	Arrangement
Single Package	HSP-A	HOSP-A	FAN INDOOR COIL / COMP OUTDOOR COIL
Remote Outdoor Coil	HRC-A-CB	HORC-A-CB	FAN INDOOR COIL COMP / OUTDOOR COIL
Remote Outdoor Coil with No Indoor Fan	HRC-A-C	HORC-A-C	INDOOR COIL COMP / OUTDOOR COIL
Split System	HRCU-A-CB	HORCU-A-CB	FAN INDOOR COIL / COMP OUTDOOR COIL
Split System with No Indoor Fan	HRCU-A-C	HORCU-A-C	INDOOR COIL / COMP OUTDOOR COIL

†A suffix of "-0" following any of the above classifications indicates equipment not intended for use with field-installed duct systems (See 5.1.3.3b)

Section 5. Testing and Rating Requirements

5.1 *Standard Ratings.* Standard ratings shall be established at the Standard Rating Conditions specified in 5.1.3. All standard ratings shall be verified by tests conducted in accordance with the ANSI/ASHRAE Standard 37-1978 *Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment* (American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc.) or ASHRAE Standard 116 *Methods of Testing for Seasonal Efficiency of Unitary Air-Conditioners and Heat Pumps,* as applicable; and with the test methods, procedures and appendixes as described in this standard.

The following types of air to air, single phase units rated less than 65,000 Btu/h cooling or less than 65,000 Btu/h heating for heating only equipment shall be tested in accordance with ASHRAE Standard 116 and rated at conditions specified in Table 3:

> SP-A; SPY-A; RCUY-A-CB; RCU-A-C; RCU-A-CB; HSP-A;
> HOSP-A; HRCU-A-CB; HORCU-A-CB: HRCU-A-C; HORCU-A-C

All other types and/or capacities of equipment within the scope of this standard shall have the capacity, EER and COP (as required) tested in accordance with ASHRAE 37-78 and rated at conditions specified in Table 4. SEER and/or HSPF may be determined on air to air equipment (as application ratings) at the option of the manufacturer.

Standard Ratings relating to cooling or heating capacities shall be net values, including the effects of circulating-fan heat, but not including supplementary heat. Power input shall be the total power input to the compressor(s) and fan(s), plus controls and other items required as part of the system for normal operation.

Standard Ratings of units which do not have indoor air-circulating fans furnished as part of the model, i.e., split system with indoor coil alone, shall be established by subtracting from the total cooling capacity 1250 Btu/h/1000 cfm [0.78 W per L/s], and by adding the same amount to the heating capacity. Total power input for both heating and cooling shall be increased by 365 W/1000 cfm [0.78 W per L/s] of indoor air circulated.

Standard Ratings of water-cooled units shall include a total allowance for cooling tower fan motor and circulating water pump motor power inputs to be added in the amount of 10 watts per 1000 Btuh cooling capacity [10 watts per 293.1 watts cooling capacity].

5.1.1 *Values of Standard Capacity Ratings.* These ratings shall be expressed only in terms of Btu/h [W] in multiples of:

Capacities Btu/h [W]	Multiples Btu/h [W]
Less than 20,000 [5,900]	100[30]
20,000 up to 38,000 [5,900 up to 11,000]	200 [60]
38,000 up to 65,000 [11,100 up to 19,000]	500 [150]
65 and above [19,000 and above]	1000 [300]

5.1.2 *Values of Measures of Energy Efficiency (EER, SEER, HSPF, and COP).* Standard measures of energy efficiency, whenever published, shall be expressed in multiples of 0.05.

5.1.3 *Standard Rating Tests.* Table 3 and Table 4 indicate the test and test conditions which are required to determine values of standard capacity ratings and values of measures of energy efficiency.

5.1.3.1 *Cyclic Tests.* Heating and cooling cyclic tests shall be conducted by cycling the compressor(s) 6 minutes "on" and 24 minutes "off". The capacity shall be measured for the integration time (θ_I) of eight (8) minutes. All electrical energy shall be measured for the integration time (θ_I) of six (6) minutes. All off cycle electrical energy shall be measured for the total cycle integration time (θ_{cyc}) of 30 minutes. Off cycle electrical energy does not include the fan power.

a. *Cooling Cyclic Test.* For units with an indoor fan, the fan power for the time period between θ_I = 6 minutes and θ_I = 8 minutes shall be added to the measured capacity.

b. *Heating Cyclic Test.* For units with an indoor fan, the fan power for the time period between θ_I = 6 minutes and θ_I = 8 minutes shall be subtracted from the measured capacity.

In lieu of conducting C and D tests or high temperature heating cycling test, an assigned value of .25 may be used for the degradation coefficient, C_D. For units with two compressor speeds, two compressors or cylinder unloading, if the assigned degradation coefficient is used for one cooling mode, it must be used for both cooling modes. If the assigned degradation coefficient is used for one heating mode, it must be used for both heating modes.

TABLE 3

Operating Conditions for Standard Rating and Performance Test Using ASHRAE Standard 116

		INDOOR UNIT		OUTDOOR UNIT	
		Air Entering		Air Entering	
	TEST	DB F	WB F	DB F	WB F
COOLING	Standard Rating Conditions "A" Cooling Steady State[1]	80	67	95	75[2]
COOLING	"B" Cooling Steady State	80	67	82	65[2]
COOLING	"C" Cooling Steady State Dry Coil	80	57[5]	82	65[2]
COOLING	"D" Cooling Cyclic Dry Coil	80	57[5]	82	65[2]
COOLING	Low Temperature Operation Cooling	67	57	67	57[2]
COOLING	Insulation Efficiency	80	75	80	75[2]
COOLING	Condensate Disposal	80	75	80	75[2]
COOLING	Maximum Operating Conditions	80	67	115	75[2]
HEATING	Standard Rating Conditions High Temperature Heating[3] Steady State	70	60 (max)	47	43
HEATING	High Temperature Heating Cyclic	70	60 (max)	47	43
HEATING	High Temperature Heating[4] Steady State	70	60 (max)	62	56.5
HEATING	Low Temperature Heating Steady State	70	60 (max)	17	15
HEATING	Frost Accumulation	70	60 (max)	35	33
HEATING	Maximum Operating Conditions	80	—	75	65

1. Same conditions used for Voltage Tolerance Tests.
2. The wet bulb temperature condition is not required when testing air-cooled condensers which do not evaporate condensate.
3. Same conditions used for Voltage Tolerance Tests (Heating-only units).
4. For two speed or two compressor units only.
5. Wet bulb sufficiently low that no condensate forms on evaporator.

TABLE 4

Operating Conditions for Standard Rating and Performance Tests Using ASHRAE Standard 37-78

		Indoor Unit		Outdoor Unit					
				Air Entering					
		Air Entering		Air Cooled		Evaporative		Water	
	TEST	DB F	WB F	DB F	WB F	DB F	WB F	IN F	OUT F
COOLING	Standard Rating Conditions, Cooling[1]	80	67	95	75[2]	95	75	85	95
COOLING	Low Temperature Operating Cooling	67	57	67	57[2]	67	57	—	70
COOLING	Insulation efficiency	80	75	80	75[2]	80	75	—	80
COOLING	Condensate Disposal	80	75	80	75[2]	80	75	—	80
COOLING	Maximum Operating Conditions	80	67	115	75[2]	100	80	90	100
HEATING	Standard Rating Conditions[3] (High temperature heating)	70	60 (max)	47	43	—	—	—	—
HEATING	Standard Rating Conditions Low Temperature Heating (Integrated)	70	60 (max)	17	15	—	—	—	—
HEATING	Maximum Operating Conditions	80	—	75	65				

5.1.3.2 *Electrical Conditions.* Recommended nameplate voltages are shown in Table 1 of ARI Standard 110 for *Air-Conditioning and Refrigerating Equipment Nameplate Voltages.*

Standarding Rating Tests shall be performed at the nameplate rated voltage and frequency.

For equipment with dual nameplate voltage ratings, Standard Rating tests shall be performed at both voltages, or at the lower of the two voltages, if only a single Standard Rating is to be published.

5.1.3.3 *Indoor-side Air Quantity.* All Standard Ratings shall be determined at an indoor-side air quantity as outlined below. All air quantities shall be expressed as cfm [L/s] of Standard Air (as defined in the ASHRAE Fundamentals Handbook).

a. Equipment with indoor fans intended for use with field installed duct systems shall be rated at the indoor-side air quantity (not to exceed 37.5 scfm per 1,000 Btu/h [0.06 L/s per watt] of rated capacity) delivered when operating against the minimum external resistance specified in 5.1.3.6 or at a lower indoor-side air quantity if so specified by the manufacturer.

b. Equipment with indoor fans not intended for use with field installed duct systems (free discharge) shall be rated at the indoor-side air quantity delivered when operating at 0 inches water [0Pa] external pressure as specified by the manufacturer.

c. Equipment which does not incorporate an indoor fan, but is rated in combination with a device employing a fan shall be rated as described under a. above. For equipment of this class which is rated for general use to be applied to a variety of heating units, the indoor-side air quantity shall be as specified by the manufacturer in his standard ratings, not to exceed 37.5 scfm/1,000 Btu/h [0.06 L/s per watt] of rated capacity or the air quantity obtained through the indoor coil assembly when the pressure drop across the indoor coil assembly and the recommended enclosures and attachment means is not greater than 0.30 inches of water [74.7 Pa], whichever is less.

Indoor-side air quantities and pressures as referred to herein apply to the air quantity experienced when the unit is cooling and dehumidfying under the Standard Rating Conditions specified in this section. This air quantity, except as noted in 5.1.3.3b and 6.4, shall be employed in all other test prescribed herein without regard to resulting external static pressure. Heating-only units shall use the air quantity experienced when the unit is operating under the High Temperature Heating Standard Rating Conditions test.

5.1.3.4 *Outdoor-side Air Quantity.* All Standard Ratings shall be determined at the outdoor-side air quantity specified by the manufacturer where the fan drive is adjustable. Where the fan drive is non-adjustable, they shall be determined at the outdoor-side air quantity inherent in the equipment when operated with all of the resistance elements associated with inlet louvers and any ductwork and attachments considered by the manufacturer as normal installation practice. Once established, the outdoor-side air circuit of the equipment shall remain unchanged throughout all tests prescribed herein.

5.1.3.5 *Requirements for Separated Assemblies.* All Standard Ratings for equipment in which the outdoor section is separated from the indoor section, as in Types RC, RCY, RCU, RCUY, HRC, HORC, HRCU and HORCU (shown in Section 4), shall be determined with at least 25 feet [7.62m] of interconnection tubing on each line, of the size recommended by the manufacturer. Such equipment in which the interconnection tubing is furnished as an integral part of the machine not recommended for cutting to length shall be tested with the complete length of tubing furnished, or with 25 feet [7.62m] of tubing, whichever is greater. At least 10 feet [3.05m] of the interconnection tubing shall be exposed to the outside conditions. The line sizes, insulation, and details of installation shall be in accordance with the manufacturer's published recommendation.

5.1.3.6 *Minimum External Pressure.* Indoor air-moving equipment intended for use with field installed duct systems shall be designed to operate against, and tested at not less than, the minimum external pressure shown in Table 5 when delivering the rated capacity and air quantity specified in 5.1.3.3. Indoor air-moving equipment not intended for use with field installed duct systems (free discharge) shall be tested at 0 in. water [0Pa] external pressure.

In interpreting this requirement, it is understood that the most restrictive filters, supplementary heating coils, and other equipment specified as part of the unit be in place and that the net external pressure specified above is available.

5.2 *Application Ratings.* Ratings at conditions of temperature or air quantity other than those specified in 5.1.3, may be published as application ratings, and shall be based on data determined by the methods prescribed in 5.1. Application ratings in the defrost region shall include net capacity and COP based upon a complete defrost cycle.

5.3 *Publication of Ratings.* Wherever application ratings are published or printed, they shall include, or be accompanied by the Standard Ratings clearly designated as such, including a statement of the conditions at which the ratings apply.

5.3.1 *Capacity Designation.* The capacity designation used in published specifications, literature or advertising, controlled by the manufacturer, for equipment rated under this standard, are to be expressed only in Btu/h [W] at the Standard Rating Conditions specified in 5.1.3 and in values prescribed in 5.1.1 and 5.1.2 horsepower, tons or other units shall not be used as capacity designation.

5.4 *Tolerances.* To comply with this standard, published cooling capacity, heat capacity(s) and efficiency ratings, shall be based on data obtained in accordance with the provisions of this section, and shall be such that any production unit, when tested, will meet these ratings except for an allowance to cover testing and manufacturing variations; the amount of allowance to be minus 5 percent.

Section 6. Performance Requirements

6.1 *Performance Requirements.* To comply with this standard, unitary equipment shall be designed and produced in accordance with the provisions of this section in such a manner that any production unit will meet the requirements detailed herein.

6.2 *Maximum Operating Conditions Test.* Unitary equipment shall be designed and produced to pass the following maximum operating conditions test with an indoor coil air quantity as determined under 5.1.3.3.

6.2.1 *Temperature Conditions.* Temperature conditions shall be maintained as shown in Table 3 or in Table 4.

6.2.2 *Voltages.* The test shall be run at the Range A minimum utilization voltage from ARI Standard 110, Table 1, based upon the unit's nameplate rated

Table 5. Minimum External Pressure

*Standard Capacity Ratings		Minimum External Resistance	
(Thousands of Btuh)	[kW]	(Inches of Water)	[Pa]
Up thru 28.0	Up thru 8.2	0.10	24.9
29 thru 42.0	8.5 thru 12.4	0.15	37.4
43 thru 70	12.6 thru 20.5	0.20	49.8
71 thru 105	20.8 thru 30.8	0.25	62.3
106 thru 134	31.1 thru 39.3	0.30	74.7

*Cooling capacity for units with cooling function; High Temperature Heating Capacity for heating-only units (See 5.1.3.3) voltage(s). This voltage shall be supplied at the unit's service connection and at rated frequency.

6.2.3 *Procedure.* The equipment shall be operated for one hour at the temperature conditions and voltage specified.

6.2.4 *Requirements.* The equipment shall operate continuously without interruption for any reason for one hour.

6.2.4.1 Units with water-cooled condensers shall be capable of operation under these maximum conditions at a water-pressure drop not to exceed 15 psi [103.4 kPa], measured across the unit.

6.3 *Voltage Tolerance Test.* Unitary equipment shall be designed and produced to pass the following voltage tolerance test with a cooling coil air quantity as determined under 5.1.3.3.

6.3.1 *Temperature Conditons.* Temperature conditions shall be maintained as shown in Table 3 or in Table 4.

6.3.2 *Voltages.*

6.3.2.1 Tests shall be run at the Range B minimum and maximum utilization voltages from ARI Standard 110, Table 1, based upon the unit's nameplate rated voltage(s). These voltages shall be supplied at the unit's service connection and at rated frequency. A lower minimum or a higher maximum voltage shall be used, if listed on the nameplate.

6.3.2.2 The power supplied to single phase equipment shall be adjusted just prior to the shut-down period (see 6.3.3.2) so that the resulting voltage at the unit's service connection is 86 percent of nameplate rated voltage when the compressor motor is on locked-rotor. (For 200V. or 208V. nameplate rated equipment the restart voltage shall be set at 179V. when the compressor motor is on locked rotor). Open circuit voltage for three-phase equipment shall not be greater than 90 percent of nameplate rated voltage.

6.3.2.3 Within one minute after the equipment has resumed continuous operation (See 6.3.4.3), the voltage shall be restored to the values specified in 6.3.2.1.

6.3.3 *Procedure*

6.3.3.1 The equipment shall be operated for one hour at the temperature conditions and voltage(s) specified.

6.3.3.2 All power to the equipment shall be cut off for a period sufficient to cause the compressor to stop (not to exceed five seconds) and then restored.

6.3.4 *Requirements.*

6.3.4.1 During both entire tests, the equipment shall operate without failure of any of its parts.

6.3.4.2 The equipment shall operate continuously without interruption for any reason for the one hour period preceding the power interruption.

6.3.4.3 The unit shall resume continuous operation within two hours of restoration of power and shall then operate continuously for one half hour. Operation and resetting of safety devices prior to establishment of continuous operation is permitted.

6.4 *Low-Temperature Operation Test (Cooling).* (Not required for heating-only units). Unitary equipment shall be designed and produced to pass the following low-temperature operation test when operating with initial air quantities as determined in 5.1.3.3 and 5.1.3.4 and with controls and dampers set to produce the maximum tendency to frost or ice the evaporator, provided such settings are not contrary to the manufacturer's instructions to the user:

6.4.1 *Temperature Conditions.* Temperature Conditions shall be maintained as shown in Table 3 or Table 4.

6.4.2 *Procedure.* The test shall be continuous with the unit on the cooling cycle, for not less than four hours after establishment of the specified temperature conditions. The unit will be permitted to start and stop under control of an automatic limit device, if provided.

6.4.3 *Requirements.*

6.4.3.1 During the entire test, the equipment shall operate without damage or failure of any of its parts.

6.4.3.2 During the entire test, the air quantity shall not drop more than 25 per cent from that determined under the Standard Rating test.

6.4.3.3 During the test, and during the defrosting period after the completion of the test, all ice or meltage must be caught and removed by the drain provisions.

6.5 *Insulation Efficiency Test (Cooling).* (Not required for heating-only units). Unitary equipment shall be designed and produced to pass the following insulation efficiency test when operating with air quantities as determined in 5.1.3.3 and 5.1.3.4 with controls, fans, dampers, and grilles set to produce the maximum tendency to sweat, provided such settings are not contrary to the manufacturer's instructions to the user:

6.5.1 *Temperature Conditions.* Temperature conditions shall be maintained as shown in Table 3 or Table 4.

6.5.2 *Procedure.* After establishment of the specified temperature conditions, the unit shall be operated continuously for a period of four hours.

6.5.3 *Requirements.* During the test, no condensed water shall drop, run, or blow off from the unit casing.

6.6 *Condensate Disposal Test (Cooling).** (Not required for heating-only units). Unitary equipment which reject condensate to the condenser air shall be designed and produced to pass the following condensate disposal test when operating with air quantities as determined in 5.1.3.3 and 5.1.3.4 and with controls and dampers set to produce condensate at the maximum rate, provided such settings are not contrary to the manufacturer's instructions to the user:

6.6.1 *Temperature Conditions.* Temperature Conditions shall be maintained as shown in Table 3 or Table 4.

6.6.2 *Procedure.* After establishment of the specified temperature conditions, the equipment shall be started with its condensate collection pan filled to the overflowing point and shall be operated continuously for four hours after the condensate level has reached equilibrium.

6.6.3 *Requirements.* During the test, there shall be no dripping, running-off, or blowing-off of moisture from the unit casing.

6.7 *Tolerances.* The conditions for the tests outlined in Section 6 are average values subject to tolerances of ± 1.0F [± 0.56°C] for air wet-bulb and dry-bulb temperatures, ± 0.5F [± 0.28°C] for water temperatures, and ± 1.0 per cent of the reading for voltages.

Section 7. Safety Recommendations

7.1 *Refrigerating System.* It is strongly recommended that the refrigerating system be designed, constructed, and assembled so as to be in accordance with the latest edition of the ANSI/ASHRAE 15-78 *Safety Code for Mechanical Refrigeration* (American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.)

7.2 *Product Safety.* It is strongly recommended that the product be designed, constructed and assembled in accordance with nationally recognized safety requirements appropriate for products covered by this standard such as the American National Standard for *Air Conditioners, Central Cooling* (ANSI/UL 465, latest edition) or American National Standard for *Heat Pumps* (ANSI/UL 559, latest edition).

7.3 *Heating Functions.* In addition to the foregoing recommendations of this section, it is strongly recommended that air-conditioners or heat pumps incorporating a gas-fired or oil-fired heating function be designed, constructed, and assembled so as to meet the applicable safety requirements of one of the following standards:

a. Gas-fired: American National Standard for *Gas-Fired Gravity and Forced Air Central Furnaces* (American National Standards Institute Standard Z21.47) or American National Standard for *Gas-Fired Heavy Duty Forced Air Heaters* (American National Standards Institute Standard Z21.53).

b. Oil-Fired: UL Standard for *Oil-Fired Central Furnaces* (Underwriters' Laboratories, Inc. Standard 727) or UL Standard for *Oil-Fired Unit Heaters* (Underwriters' Laboratories, Inc. Standard 731).

Section 8. Conformance

8.1 *Conformance.* While conformance with this standard is voluntary, conformance shall not be claimed or implied for products or equipment within its Purpose (Section 1) and Scope (Section 2) unless such claims meet all of the requirements of the Standard.

*This test may be run concurrently with the Insulation Efficiency Test (See 6.5).

APPENDIX A

Bin Hours Cooling and Heating

1. The following values shall be used in conjunction with ASHRAE 116 when calculating Seasonal Performance Factors (SEER, HSPF) for standard rating purposes.

1.1 Distribution of cooling hours in temperature bins to be used for calculation of SEER based on representative use cycle of 1000 cooling hours per year.

Bin Number, j	Bin Temperature, T_j(°F)	Bin Hours n_j
1	67	214
2	72	231
3	77	216
4	82	161
5	87	104
6	92	052
7	97	018
8	102	004

1.2 Cooling outdoor design temperature (t_{OD}) shall be 95°F.

1.3 Cooling building load Size Factor shall be 1.1 for 10% oversizing.

1.4 Distribution of heating hours in temperature bins to be used for calculation of HSPF based on representative use cycle of 2250 heating hours per year (Region IV).

Bin Number j	Bin Temperature T_j(°F)	Bin hours n_j
1	62	297
2	57	250
3	52	232
4	47	209
5	42	225
6	37	245
7	32	283
8	27	196
9	22	124
10	17	81
11	12	58
12	7	29
13	2	13
14	−3	4
15	−8	2

APPENDIX B

Rating Equipment with Variable Speed Compressors

Test conditions and methodology for rating unitary air-conditioning and heat pumps with variable speed compressors as diagrammed in Figures 1 and 2.

1. *Test Conditions*

1.1 The tests and test conditions which are required to determine values of capacity ratings and values of measure of energy efficiency appear in Table B1.

1.2 In lieu of conducting the Cyclic Cooling and Low Ambient Dry Coil tests, an assigned value of .25 may be used for the cooling degradation coefficient, C_D.

1.3 In lieu of conducting the Cyclic Heating test, an assigned value of .25 may be used for the heating degradation coefficient, C_D.

2. *Calculations* (NOTE: Refer to Bin Analysis method in ASHRAE 116)

2.1 *Cooling Seasonal Performance Factor.* For variable speed compressor units the seasonal energy efficiency ratio (SEER) is found from the following equation:

$$SEER = \frac{\sum_{j=1}^{8} q(t_j)}{\sum_{j=1}^{8} E(t_j)}$$

The terms $q(t_j)$ and $E(t_j)$ as defined in ASHRAE 116, summed over temperature bins, are evaluated at each temperature bin according to the three possible cases listed below.

The building cooling load $BL(t_j)$ for the three cases is obtained from the following equation:

eq.(1)

$$BL(t_j) = \frac{t_j - 65}{95 - 65} \frac{[q_{ss}^{k=2}(95F)]}{\text{Size Factor}}$$

where $q_{ss}^{k=2}(95F)$ = Steady state capacity measured from Test "A".

Size Factor = 1.1 for 10% oversizing.

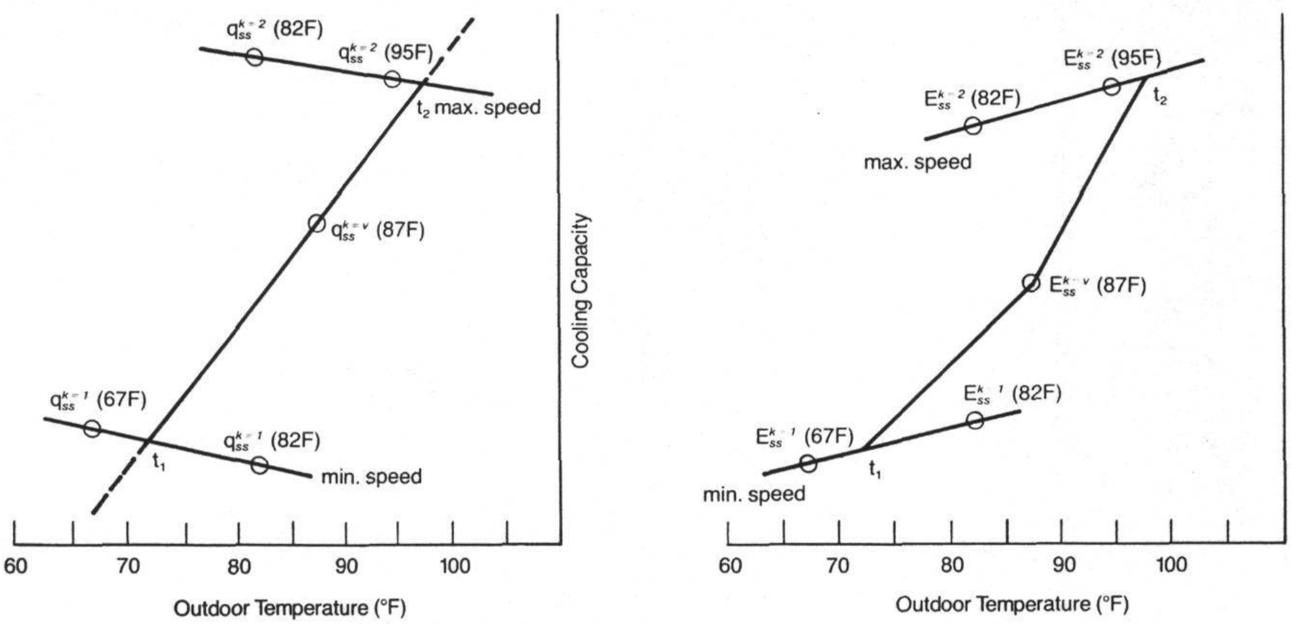

Figure 1. Diagram of Test Data Points For Unitary Air Conditioners And Heat Pumps With Variable Speed Compressors in the Cooling Mode.